



**National Aeronautics and
Space Administration**

George C. Marshall Space Flight Center
Science and Engineering
Contract NAS 8-37814

MECHANICAL FLEXIBLE JOINT

DESIGN DOCUMENT

Report No.: 324-005-93-041

Prepared By:

Vic Daily

October 22, 1993

(NASA-CR-193941) MECHANICAL
FLEXIBLE JOINT DESIGN DOCUMENT
(Sverdrup Technology) 108 p

N94-28419

Unclass

G3/37 0209831

Sverdrup
CORPORATION

Sverdrup Technology, Inc.
MSFC Group
620 Discovery Drive
Huntsville, Alabama 35806

PREFACE

This report was compiled for EP42 under contract NAS-8-37814, Task 324-005, Subtask 01.

The purpose of the report is to document the status of the Mechanical Flexible Joint Design Subtask with the intent of halting work on the design. Recommendations for future work is included in the case that the task is to be resumed.

TABLE OF CONTENTS

Page

TABLE OF CONTENTS		i
LIST OF FIGURES		iii
1.0 INTRODUCTION		1
2.0 MECHANICAL FLEXIBLE JOINT (MFJ)		2
2.1 MFJ DESIGN REQUIREMENTS		2
2.2 MFJ DESIGN ISSUES TO BE ADDRESSED		2
2.3 MFJ DESIGN CONSIDERATIONS.....		3
2.3.1 Two Approaches for Designing a MFJ		3
2.3.2 Sealing Surface vs Bearing Surface		4
2.3.3 Self - Centering of the MFJ		4
2.4 ANALYTICAL DESIGN OF A MFJ		4
2.4.1 Ball Radius Equations.....		5
2.4.1.1 Typical Ball Equation		6
2.4.1.2 Concentric Single Ball Equation.....		7
2.4.1.3 Concentric Dual Ball Equation.....		8
2.4.2 Torque Equations.....		9
2.4.2.1 Force Equations		9
2.4.2.2 Rotational Torque Equations.....		10
2.4.2.3 Hinge Torque Equation.....		11
2.4.3 Weight		12
2.5 MFJ CONCEPTS.....		12
2.5.1 Concept 1 Two Sides / Single Pivot Point.....		12
2.5.2 Concept 2 Two Sides / Dual Pivot Point.....		12
2.5.3 Concept 3 One Side / Single Pivot Point.....		12
2.5.4 Concept 4 Two Sides / Gimbal Ring		13
2.5.5 Concept 5 Two Sides / Concentric Bearing.....		13
2.5.6 Securamax Inc.		13
2.5.7 Stainless Steel Products Inc.		13
2.6 TRADE STUDY		13
3.0 MFJ LITERATURE / HISTORICAL SEARCH		15
3.1 SSME LOW PRESSURE FUEL DUCT CONFIGURATION		15
3.2 LITERARY / HISTORICAL SEARCH OF FLEX JOINT DESIGNS		16
3.3 COMMERCIAL VENDORS OF FLEX JOINTS.....		17
3.4 PERSONAL EXPERIENCE DESIGNING BELLOWS / FLEX JOINTS.....		20
4.0 SEALS		23
4.1 DESIGN VARIABLES FOR SEALING		23
4.2 DSTF SEAL VENDORS		24
4.3 SEAL EVALUATION DATA.....		24
4.3.1 Seal Vendor Specifications		24
4.3.2 Seal Vendor Prices		24
4.3.3 Seal Vendor Materials.....		25
4.4.4 More Specifics of Seal Vendors		25

	Page
5.0 DYNAMIC SEALS TEST FIXTURE (DSTF)	27
5.1 DSTF DESIGN ISSUES TO BE ADDRESSED	27
5.2 HOW DOES DSTF WORK?	28
5.3 MATERIALS SELECTION	28
5.4 DESIGN CONSIDERATIONS FOR DYNAMIC CRYOGENIC SEALS	29
5.5 STRUCTURAL ANALYSIS	30
5.5.1 Stress Analysis	30
5.5.2 Thermal Analysis	30
5.5.3 Weld Fatigue Analysis	31
5.6 CHANGES IN THE ORIGINAL DSTF STRATEGY	31
5.6.1 Controlled Environment	31
5.6.1.1 Heaters	31
5.6.1.2 Single vs. Multiple Test Sleeves	31
5.7 DSTF PDR DRAWINGS	32
5.8 PROCUREMENT OF DSTF	32
5.9 MODIFIED DSTF	33
6.0 BEARING AREA COATINGS	34
6.1 DIAMOND FILM PROCESS FOR MFJ APPLICATIONS	34
6.1.1 Summary	34
6.1.2 Introduction	34
6.1.3 Film Deposition Processes	35
6.1.3.1 CVD Process	35
6.1.3.2 IBED Process	35
6.1.3.3 PACVD Process	35
6.1.4 Other Low Friction Films	36
7.0 INSULATION	38
7.1 MECAHNICAL FLEX JOINT	38
7.2 DYNAMIC SEALS TEST FIXTURE	38
8.0 TEST PLAN	39
8.1 DYNAMIC SEALS TEST PLAN	39
8.2 MECHANICAL FLEX JOINT TEST PLAN	39
9.0 CONCLUSION	40
10.0 RECOMMENDATIONS	41

APPENDICES:

Appendix A: MFJ Design Requirements Document

Appendix B: Static Seals Test Plan

Appendix C: Torque & Ball Radius Equations Program

Appendix D: DSTF PDR Drawing Package (Seperate binding)

LIST OF FIGURES:

FIGURE 1 Envelope Comparison between MFJ Concept 5 and Bellows

FIGURE 2 Sealing Surface vs. Bearing Surface

FIGURE 3 Concept 1 of MFJ - Two Sides/Single Pivot

FIGURE 4 Concept 2 of MFJ - Two Sides/Dual Pivot

FIGURE 5 Concept 3 of MFJ - One Side/Dual Pivot

FIGURE 6 Concept 4 of MFJ - Two Side/Gimbal Ring

FIGURE 7 Concept 5 of MFJ - Two Side/Concentric Bearing

FIGURE 8 Concept 6 of MFJ - Securamax

FIGURE 9 Concept 7 of MFJ - Stainless Steel Products

FIGURE 10 Mechanical Flex Joint Ranking

FIGURE 11 Modified Dynamic Seals Test Fixture

FIGURE 12 Typical MFJ

FIGURE 13 Compact MFJ

FIGURE 14 Typical MFJ Bearing Forces

1.0 INTRODUCTION

The Mechanical Flex Joint (MFJ) is designed to replace the Short Flex Joint (see Section 3.1, P/N RS008981) on the Space Shuttle Main Engine (SSME) Low Pressure Fuel Duct.

1.1 WHAT DOES A FLEX JOINT DO?

The Low Pressure Fuel Duct (see Section 3.1, P/N RS007018) is an assembly of a 5.2 inch I.D. cryogenic fuel line sections and three flex joints that conveys LH2 to the High Pressure Fuel Turbopump (HPFTP). When this fuel duct feeds LH2 to the HPFTP the duct shrinks due to thermal contraction and tends to straighten out any bends. The flex joints allow the duct to accommodate those thermal deflections, flange misalignments and the necessary gimbling without bending or crimping. The flex joints have two degrees of freedom; therefore, twisting the flex joint is not possible.

1.2 WHY REPLACE THE FLEX JOINTS?

The current flex joint has a ball and socket, or "tripod configuration" (see P/N RS008961, Sheet 6), in the center to take axial loads and pressure thrust. The flex joints have experienced cracking in the "tripod configuration" which is caused by undersized tripod leg radii creating a localized over stressed area, which combined with high cycle fatigue could cause the flex joint leg to crack and eventually fail. If the cracking caused part of the flex joint to "break up" and be sucked into the HPFTP during flight, the results could be catastrophic. Another problem, with the current flex joints, is the possibility of fatigue cracks in the bellows or the outer containment jackets. Fatigue cracking can be caused by high cycle fatigue. Fatigue cracks have never been a problem before but can occur over time. A failure of the jackets could also be catastrophic.

1.3 WHY A MECHANICAL FLEX JOINT?

The MFJ is designed to eliminate two failure points from the current flex joint configuration, the inner "tripod configuration" and the outer containment jacket. The MFJ will also be designed to flex 13.5 degrees and have three degrees of freedom. By having three degrees of freedom the, MFJ will allow the Low Pressure Fuel Duct to twist and remove the necessity to angulate the full 11 degrees currently required.

The current flex joints are very labor intensive and very costly and a simple alternative is being sought. The MFJ is designed with a greater angular displacement, with three degrees of freedom, to reside in the same overall envelope, to meet weight constraints of the current bellows, to be compatible with cryogenic fuel and oxidizers, and also to be man-rated.

2.0 MECHANICAL FLEX JOINT

The following section has been compiled to provide our "lessons learned" gained by designing the Mechanical Flex Joint (MFJ). The following tips are abbreviated and therefore require further investigation of each of them. The first sub-section listed is a review of the MFJ design requirements. A copy of the entire MFJ Design Requirements Document is included in Appendix A of this report.

2.1 MFJ DESIGN REQUIREMENTS:

Design Temperature: -415 degrees F
-Liquid Nitrogen Temperature: -320 degrees F
-Liquid Hydrogen Temperature: -423 degrees F
-Liquid Helium Temperature: -452 degrees F
Maximum Design Pressure: 343 psia
Maximum Operating Pressure: 326 psia
Calculated Burst Pressure: 610 psig @ -305 degrees F
Design Angular Deflection: +/- 13.5 degrees omni-directional
Cycle Life: 400 full angular cycles @ 343 psia and -415 degrees F
2800 non-operational @ ambient temperature
Deflection Torque: 7355 in-lb @ 11.5 degrees @ 326 psia and -415 degrees F
5019 in-lb @ 11.5 degrees @ ambient pressure and temperature
Structural Loads adapted from Feed line Spec. RSS-8561-24 at Worst Case for Flex Joint location
Duct Size: 5.20 in. ID Wall Thickness: 0.032 in.
External Leakage: Shall not exceed 1×10^{-3} scc/sec GHe @ design pressure & temp.
Safety factor @ yield is 1.26
Safety factor @ ultimate is 1.81
Redundant Seals as requirement for man-rating

2.2 MFJ DESIGN ISSUES TO BE ADDRESSED:

- a.) Movable cryogenic seal
- b.) Torque
- c.) Thermal shock/Deflection
- d.) Dynamic Loading
- e.) Fabrication
- f.) Reliability
- g.) Weight
- h.) Stress analysis
- i.) Pressure drop
- j.) Movable cryogenic insulation

Although each of these issues is critical to designing a MFJ, the opportunity to address all of them was not accomplished before this effort was terminated.

2.3 MFJ DESIGN CONSIDERATIONS

There were a number of variables to consider when designing a MFJ. Some of these design considerations can be controlled by engineering judgment, i.e. diameter and length while other variables have to be approached from an analytical standpoint i.e. rotational and hinge torque. The ball radius, on the other hand, can be controlled by either the designer or a ball radius equation. This ball diameter in turn can control the diameter, length, weight and torque of the MFJ. As one can see, the design process can become very complicated. Below is a simplified definition of some design considerations that are used to design the MFJ.

The original MFJ flex angle is designed using an angular deflection of 11 degrees, was changed to 13 degrees, and finalized at 13.5 degrees. The five conceptual designs were all conceived using the 13 degrees instead of 13.5 degrees. These conceptual designs will require minimal modification to adhere to the 13.5 degrees of angular deflection and any dimensional changes can be calculated using the equations provided in Sections 2.4.

2.3.1 Advantages/Disadvantages of a MFJ

Sverdrup presented five conceptual designs to NASA's MSFC EP64 Branch. These designs combine a variety of ideas. The following are the two approaches for designing a MFJ and the typical advantages and disadvantages:

Two Sides Pivot - each end pivots 6.5 degrees, half the requirement. (See Figure 3)

Advantages: Ball diameter is smaller.

Disadvantages: Two sets of redundant seals - two leak paths or failure points.

One Side Pivots - one end pivots 13 degrees (see Figure 5)

Advantages: One set of redundant seals - one leak path or failure point.

Disadvantages: Ball diameter is typically larger.

Below is listed the advantages and disadvantages of flex joints as compared to each of the five designs.

Other Advantages/Disadvantages of MFJ Compared to Current Flex Joints

	<u>Advantage</u>	<u>Disadvantage</u>
Weight:	≤ 50 lbs.	> 50 lbs.
Torque:	≤ 7355 ft-lbs.	>7355 ft-lbs.
Diameter:	≤ 8.83 in.	> 8.30 in.
Length:	≤ 8.70 in.	> 8.70 in.
Flow Liner:	YES	
Rotational:	YES	

As the angular deflection and/or bearing area increases so does the ball diameter and length. If the length and diameter push the flex joint out of the current envelope of the bellows (See Figure 1) the feasibility of building the flex joint diminishes. Sverdrup's 13 degree angulation designs has a ball/pipe diameter ratio of approximately 2:1. i.e. a 5 inch diameter duct has a 10 inch diameter ball. Refer to Section 2.4.1, Ball Radius Equation, for specifics regarding this subject.

2.3.2 Sealing Surface vs. Load Bearing Surface

When designing this joint, a sealing and a load bearing area cannot share common surfaces. This is not desirable because the bearing area could mar or scratch the sealing surface. If the sealing surface is damaged, it could cause damage to the sealing jacket material resulting in failure of the seals and, therefore, failure of the MFJ. Figure 2 illustrates the concept of how the sealing and bearing areas cannot share coincident surfaces.

Section 6, Bearing Surfaces, will address different surface materials for MFJ designs.

2.3.3 Self-Centering MFJ

One design consideration not to be overlooked is the requirement that the MFJ self-center while in a static environment. Currently the bellows configuration outer jacket has the rigidity to maintain a center position when not in use. The MFJ, on the other hand, lays "limp" when not in use. This lack of alignment, or preload, can be a major problem when considering the overall Low Pressure Fuel Duct configuration and assembly process.

RECOMMENDATION: This issue deserves further investigation.

2.4 ANALYTICAL DESIGN OF A MFJ

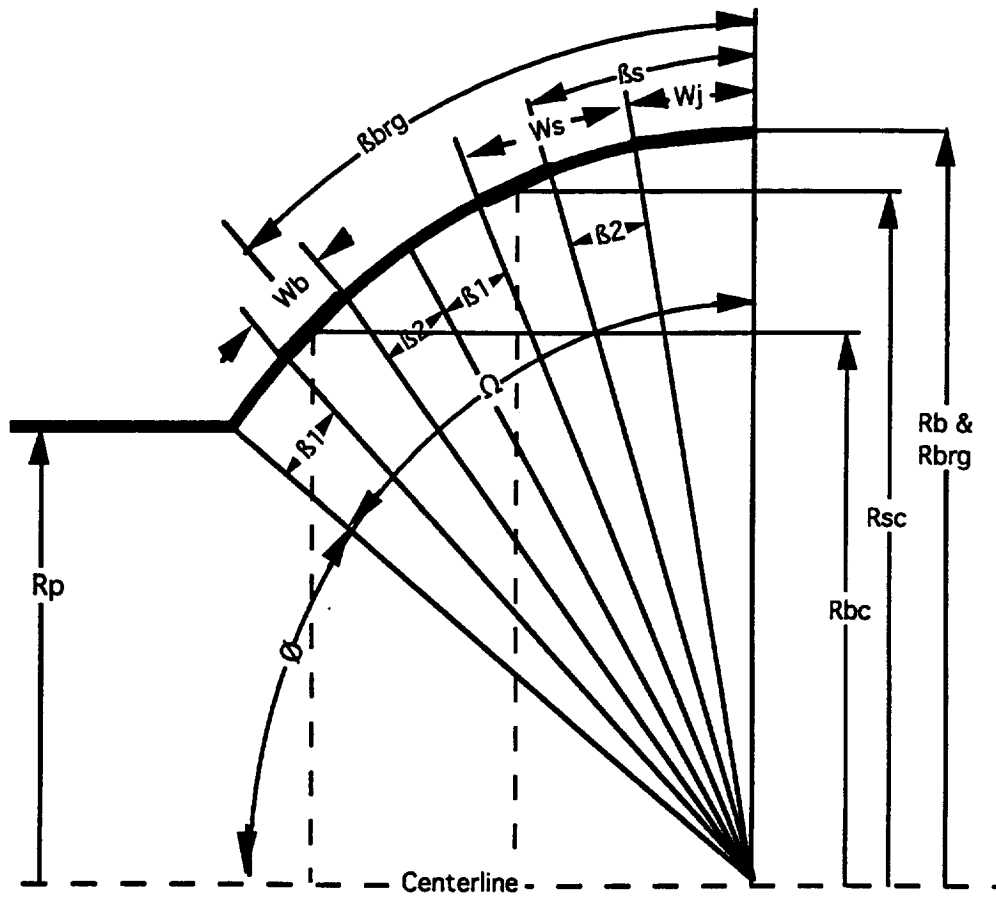
Each design concept has been ranked according to ranking criteria established with EP64 prior to this study (See Figure 10). Equations derived to express ranking criteria provide relative performance and trends of the selected concepts. Strong and weak points of each MFJ concept are discussed and presented in Section 2.5, MFJ Concepts.

The size of a MFJ may be calculated and allowances made to determine minimum diameter and weight. The ball-and-socket concept pursued has identified several improvements to a typical baseline design. Elements of a typical MFJ design that effect size are: seal width, bearing width, joint width and flex angle (See Figure 2). In addition, the seal width must also include seal containment width and edge distance. These elements, located on a spherical surface, combine with the required pipe size to define a 90 degree arc length that is the spherical working surface of the MFJ. The minimum ball radius, R_b , provides the minimum arc length necessary to contain all the design elements. The pipe inside diameter is the design point of beginning. The seal diameter should be located as near to the pipe as possible (minimum diameter) to minimize internal pressure loads contained within the structure. The bearing surface should also be located as far away from the ball equator as possible because the bearing angle, β_{brg} , produces a wedge to contain the pressure loads. A bearing angle near 45 degrees is desirable to balance hoop stress and axial stress.

Two design improvements to reduce MFJ size are presented. A concentric bearing, Figure 7, utilizes the same arc length for both bearing and seal and also locates the bearing further away from the equator. Another improvement, a dual ball, Figures 3 and 4, reduces the flex angle by flexing both ends half the required angle.

The following equations predict the spherical ball sizes of these three MFJ options. This ball diameter is the major influence on MFJ outside diameter. Allowance for structural thickness and attachment joint must also be included in the over-all diameter.

2.4.1 Ball radius equations



TYPICAL MFJ

FIGURE 12 TYPICAL MFJ

Definition of Symbols

$\frac{R_b}{R_p}$ = Ball/Pipe size ratio	W_b = Arc length of bearing, in
R_p = Pipe inside radius, in	W_s = Arc length of seal, in
R_b = Ball radius, in	W_j = Arc length of joint, in
ϕ = Arc produced by pipe radius, radians	β_{brg} = Bearing contract angle, deg.
Ω = Arc length of ball, radians	R_{bc} = Bearing Cord Radius, in
β_1 = Positive flex angle, degrees	β_s = Seal contact angle, deg.
β_2 = Negative flex angle, degrees	R_{sc} = Seal Cord Radius, in

2.4.1.1 Typical Ball Equation:

$$\Omega = \frac{\pi}{2} - \phi$$

$$\phi = \arcsin \frac{R_p}{R_b}$$

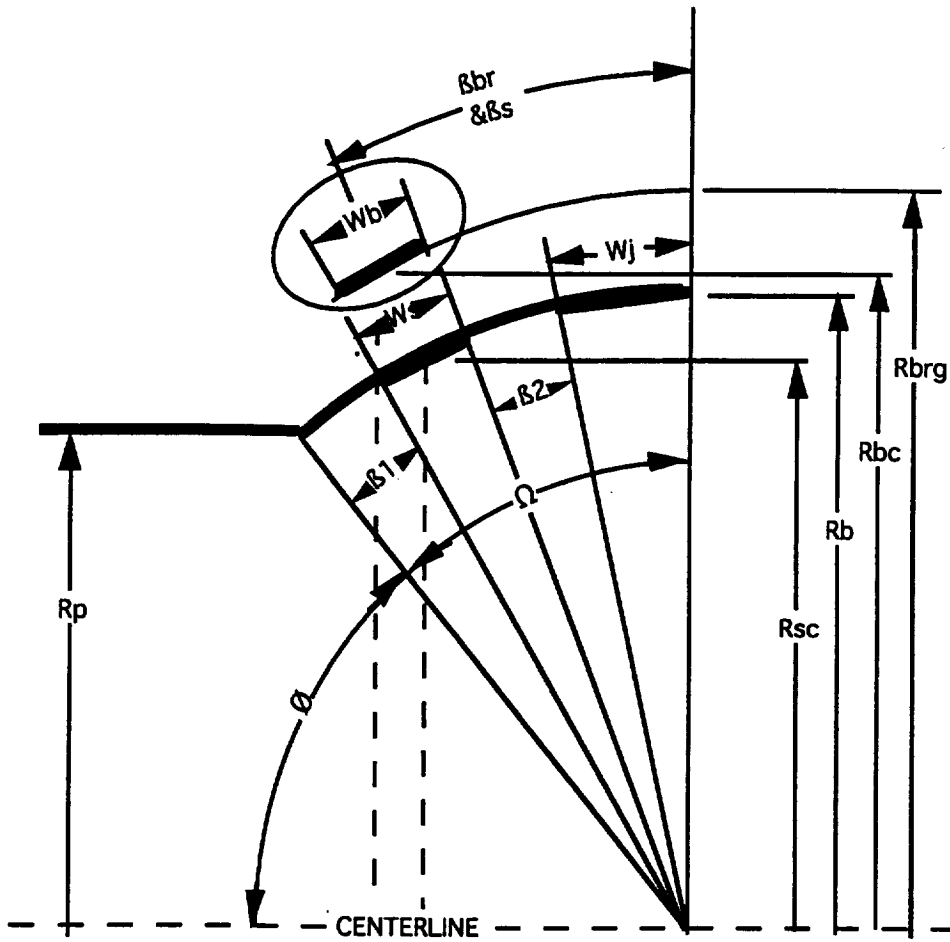
$$\beta_{brg} = \frac{W_b}{2R_{brg}} + \frac{W_j}{R_b} + \frac{W_s}{R_b} + 2\beta_2 + \beta_1$$

$$\beta_s = \frac{W_s}{2R_b} + \frac{W_j}{R_b} + \beta_2$$

$$(Eq. 1) \quad R_b \Omega = R_b \left(\frac{\beta_1 + \beta_2}{57.2958} \right) (2) + W_s + W_b + W_j$$

$\frac{R_b}{R_p}$ = iterative solution of Equation 1 from computer program, Appendix C.

$\frac{R_b}{R_p} = 3.75$ is the Typical Ball Concept ratio.



COMPACT MFJ

FIGURE 13 COMPACT MFJ

Definition of Symbols:

$\frac{R_b}{R_p}$ = Ball/Pipe size ratio	W_b = Arc length of bearing, in
R_p = Pipe inside radius, in	W_s = Arc length of seal, in
R_b = Ball radius, in	W_j = Arc length of joint, in
ϕ = Arc produced by pipe radius, radians	β_{brg} = Bearing contract angle, deg.
Ω = Arc length of ball, radians	R_{bc} = Bearing Cord Radius, in
β_1 = Positive flex angle, degrees	β_s = Seal contact angle, deg.
β_2 = Negative flex angle, degrees	R_{sc} = Seal Cord Radius, in

2.4.1.2 Concentric Single Ball Equation:

$$\Omega = \frac{\pi}{2} - \phi$$

$$\phi = \arcsin \frac{R_p}{R_b}$$

$$\beta_{brg} = \frac{W_b}{2R_{brg}} + \frac{W_j}{R_b} + \beta_2$$

$$\beta_s = \frac{W_s}{2R_b} + \frac{W_j}{R_b} + \beta_2$$

$$(Eq. 2) \quad R_b \Omega = R_b \left(\frac{\beta_1 + \beta_2}{57.2958} \right) + W_s + W_j$$

$\frac{R_b}{R_p}$ = iterative solution - modified program, Appendix C.

$\frac{R_b}{R_p} = 1.84$ is the Compact MFJ Concept ratio for single concentric bearing.

2.4.1.3 Concentric Dual Ball Equation:

Dual ball rotates both ends and reduces size by flexing both ends of the mechanical flexible joint. The required flex angles are effectively reduced in half in these equations.

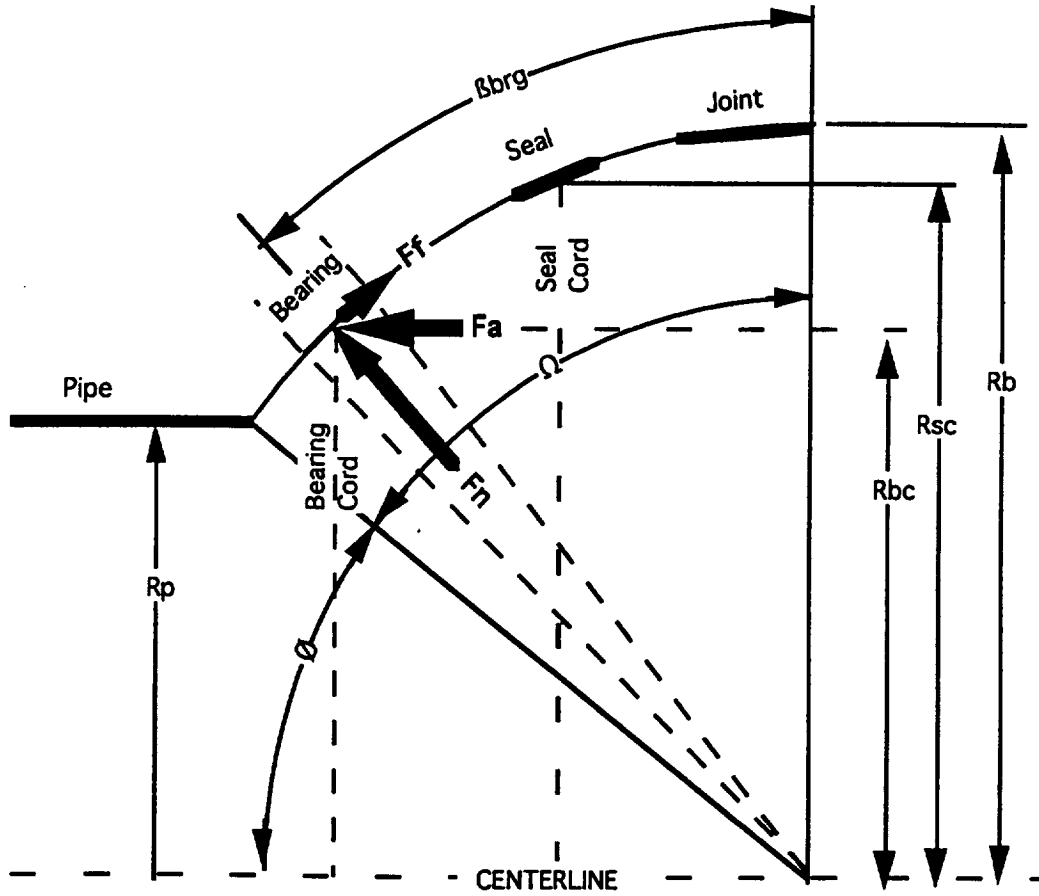
$$(Eq. 3) \quad R_b \Omega = R_b \left(\frac{\beta_1 + \beta_2}{(2) 57.2958} \right) + W_s + W_j$$

$\frac{R_b}{R_p}$ = iterative solution - modified program, Appendix C.

$\frac{R_b}{R_p} = 1.54$ is the Compact MFJ Concept ratio for dual concentric bearing.

With the selected concentric dual ball concept, the MFJ ball is only 54% larger than the pipe diameter. This concept resulted in minimum ball diameter and is therefore judged to be the best design approach for a Mechanical Flex Joint.

2.4.2 Torque Equations



TYPICAL MFJ BEARING FORCES

FIGURE 14 TYPICAL MFJ BEARING FORCES

2.4.2.1 Force Equations

$$F_a = P_d A_{sc}$$

$$A_{sc} = \pi(R_{sc})^2$$

$$\text{Eq. (4)} \quad F_a = P_d \pi(R_{sc})^2$$

$$\text{Eq. (5)} \quad F_n = \frac{F_a}{\sin \beta_{brg}}$$

$$F_f = \mu F_n$$

$$\text{Eq. (6)} \quad F_f = \frac{\mu F_a}{\sin(\beta_{brg})}$$

Where:

- F_a = Axial force separating MFJ, lb
- F_n = Normal force (perpendicular) to bearing surface, lb
- F_f = Friction force tangent to bearing surface, lb
- μ = Coef friction
- R_{sc} = Radius of Seal Cord, in
- β_{brg} = Bearing contact angle, deg.
- P_d = Design operating pressure, psia

Rotation Torque Equations

Torque necessary to rotate a mechanical flexible joint around the axial centerline (pipe centerline) can be calculated from friction on the spherical bearing surface produced by internal pressure loads. Torque is reduced with lower design pressure, smaller seal diameter, and larger bearing angle. Friction coefficient directly effects torque; and lubricants are limited, in this application, to those materials and processes compatible with cryogenic rocket fuels and associated thermal shock.

2.4.2.2 Rotation Torque Equation

- T_R - Rotation Torque, ft-lbf
- R_{bc} - Bearing Cord Radius, in
- F_f - Friction Force, lbf
- μ - Coefficient of friction
- F_n - Bearing contact force, lbf
- F_a - Axial Force, lbf
- β_{brg} - Bearing contract angle, degrees
- P_d - Design pressure, psia
- A_s - Pressure Area bounded by seal, in²
- R_s - Radius of seal, in

$$(\text{Eq. 1}) \quad T_R = \frac{F_f R_{bc}}{12}$$

$$R_{bc} = R_b \cos \beta_{brg}$$

Substituting in (Eq. 1)

$$T_R = \mu \left[\frac{(P_d) \pi (R_{sc})^2}{\sin(\beta_{brg})} \right] \left[\frac{(R_b) \cos(\beta_{brg})}{12} \right]$$

$$(Eq. 2) \quad T_R = \frac{\mu R_b P_d (R_{sc})^2}{3.8197 \tan(\beta_{brg})}$$

2.4.2.3 Hinge Torque Equation

Torque necessary to rotate a mechanical flexible joint around the spherical pivot point can be calculated from friction on the spherical bearing surface produced by internal pressure loads. Hinge torque is reduced with lower design pressure, smaller seal diameter, and larger bearing angle. Friction coefficient directly effects torque; and lubricants are limited, in this application, to those materials and processes compatible with cryogenic rocket fuels and associated thermal shock.

- T_F - Flex Torque, ft-lbf
- F_f - Friction Force, lbf
- R_{cg} - Radius to center of gravity, in
- R_{brg} - Bearing Radius, in
- μ - Coefficient of friction
- F_n - Bearing contact force, lbf
- F_a - Axial Force, lbf
- β_{brg} - Bearing contract angle, degrees
- P_d - Design pressure, psia
- A_s - Pressure Area bounded by seal, in²
- R_s - Radius of seal, in

$$(Eq. 5) \quad T_F = \frac{F_f R_{cg}}{12}$$

$$R_{cg} = 0.4244 R_{brg}$$

$$F_f = \mu F_N$$

$$F_n = \frac{F_a}{\sin(\beta_{brg})}$$

$$F_a = P_d A_s$$

$$A_s = \pi (R_{sc})^2$$

Substituting in (Eq. 5):

$$T_F = \mu \left(\frac{P_d \pi (R_{sc})^2}{\sin(\beta_{brg})} \right) \left(\frac{0.4244 R_{brg}}{12} \right)$$

$$(Eq. 6) \quad T_F = \frac{\mu R_{brg} P_d (R_{sc})^2}{9 \sin(\beta_{brg})}$$

2.4.3 Weight

The weight of each concept was calculated by determining the its' volume and multiplying it by the weight of the material to be used. At the time this trade study was conducted the material was Inconel 625.

2.5 MFJ CONCEPTS

Seal suppliers which were contacted could only identify one MFJ design that is used and it is a high temperature, low pressure jet engine exhaust duct. Two basic configurations of a MFJ were studied; ball & socket and ball & gimbal ring. Alternate design concepts were narrowed to five independent concepts. Two additional MFJ concepts were received from potential suppliers of space flight hardware. The selected designs are presented in Figure 3 through Figure 9. Evaluation of these seven design concepts follow in next section. The two vendor concepts (Figure 8 & 9) can be cross-referenced in Section 3, Literary / Historical Search.

2.5.1 Concept 1 - Two Sides/Single Pivot, (See Figure 3), Ball radius-6.279 in. Ball Length-11.463 in. Bearing Angle-45 degrees Torque-41.207 in-lbs. Weight-52.64 lbs.

The spherical shape resulted in both a large diameter and added length. Both ends articulate and thus the arc length is half (single pivot arc length) and the diameter is reduced; however, a second pair of seals is necessary. While operating pressure exerts force in a direction to close this structure, separate external flanges are necessary at both ends to maintain closure and seal contact when unpressurized. This ball concept offers three degrees of freedom, which allows the ability to rotate and thus reduces the number of joints and actually necessitates fewer degrees of deflection. The predicted weight is the highest of the five concept designs presented. Torque is the second highest due to large diameter (second largest), which increases seal diameter (pressure area), torque arm, and length. This hollow structure has no inherent flow directing baffles and is expected to exhibit more turbulent flow than other concepts. Overall, Concept 1 has been judged a poor performer and ranks last. See Figure 10 for the overall ranking of Concept 1 in the MFJ Trade Study.

2.5.2 Concept 2 - Two Sides/Dual Pivot, (See Figure 4), Ball radius-5.7 in. Ball Length-6.2 inch Bearing Angle-39 degrees Torque-30.206 in-lbs. Weight-30.4 lbs.

The spherical shape is truncated and dual pivot points overlap to reduce length. Both ends articulate, thus arc length and diameter are reduced; but design still requires two set of seals. While operating pressure exerts force in a direction to close this structure, a separate external flange is necessary at both ends to maintain closure and seal contact when unpressurized. This ball concept offers three degrees of freedom which allow the ability to rotate, and thus reduces the number of joints and actually necessitates fewer degrees of deflection. Predicted weight and torque is reduced. This shortened hollow structure has no inherent flow directing baffles and is expected to exhibit more turbulent flow than some other concepts. Overall, Concept 2 has been judged a marginal performer. See Figure 10 for the overall ranking in the MFJ Trade Study.

2.5.3 Concept 3 - One Side/Single Pivot, (See Figure 5), Ball radius-6.4 inch Ball Length-9.3 inch Bearing Angle-46 degrees Torque-43.653 in-lbs Weight-38.9 lbs.

A hemispherical movable end is joined to a fixed conical end. The minimum (one pair) number of seals are necessary for this simple concept. Only one end articulates: thus arc length and diameter of the movable end are maximum size to meet flex angle requirements. While operating pressure exerts force in a direction to close this structure, a separate external flange is necessary to maintain closure and seal contact when unpressurized. This

ball concept offers three degrees of freedom, which allow the ability to rotate, and thus reduces the number of joints and actually necessitates fewer degrees of deflection. Predicted weight is between Concept 1 and 2. This hollow structure has no inherent flow directing baffles and is expected to exhibit more turbulent flow than some other concepts. Overall, Concept 3 has been judged a poor performer. See Figure 10 for the overall ranking of Concept 3 in the MFJ Trade Study.

2.5.4 Concept 4 - Two Sides/Gimbal Ring, (See Figure 6), Ball radius-6.8 inch Ball Length-13.3 inch Bearing Angle-52 degrees Torque-2410 in-lbs. Weight-110.2 lbs. This concept is hemispherical on both ends; and furthermore, both ends must flex the full 13 degrees independently of each other. Therefore, this is the concept with the largest diameter, the longest and is the heaviest design considered. While operating pressure exerts force in a direction to close this structure, a separate external flange is necessary to maintain closure and seal contact when unpressurized. This gimbal concept has only two degrees of freedom and will not permit rotation to accommodate feed line deflection. Because of this constraint, the feed line may have to accommodate more gimbal ring joints, much like the current bellow configuration. Weight can be reduced somewhat by using hollow flow baffles; however, this issue has not been pursued nor considered worthwhile due to the size constraints discussed. Torque is very low due to both low friction bearing material and small torque arm (bearing radius). This structure has inherent flow directing baffles and has been expected to exhibit less turbulent flow than some other concepts. Overall, Concept 4 was rated the worst performer for flight hardware. See Figure 10 for the overall ranking of Concept 4 in the MFJ Trade Study.

2.5.5 Concept 5 - Two Sides/Concentric Bearing, (See Figure 7), Ball radius-4.3 inch Ball Length-5.9 inch Bearing Angle-64 degrees Torque-19875 in-lbs. Weight-18.9 lbs.

The spherical shape resulted in both small diameter and length. Both ends articulate, thus arc length is half (single pivot arc length), and diameter is reduced. Necessity for a second pair of seals has been eliminated by extending one spherical surface to overlap the other and sealing between them. A baffle has been attached to the outlet end to reduce turbulent flow. Arc length was further reduced by locating one bearing surface outboard and concentric to the seal. Arc length has been further reduced by floating both bearings (between stops) to permit independent articulation of two sides, although a single pivot point resulted. This concept also eliminated need for two external flanges to maintain seal contact. This ball concept offers three degrees of freedom which allow the ability to rotate thus reducing the number of joints and actually necessitating fewer degrees of deflection. Predicted weight is much less than any other design presented. Overall, Concept 5 was ranked the best performer. See Figure 10 for the overall ranking in the MFJ Trade Study.

2.5.6 Securamax, See Figure 8 for a conceptual sketch and Figure 10 for the overall ranking of the Securamax design in the MFJ Trade Study. For additional information regarding this design, refer to Securamax in Section 3, the Literary/Historical Search.

2.5.7 Stainless Steel Products, See Figure 9 for a conceptual sketch and Figure 10 for the overall ranking Stainless Steel Products design in the MFJ Trade Study. For additional information regarding this design, refer to Stainless Steel Products in Section 3, the Literary/Historical Search.

2.6 TRADE STUDY

MFJ ranking criteria have been evaluated for performance impact in selection of candidate concepts to meet design requirements. The trade study approach emphasized that

a primary goal is to reduce diameter and length of the spherical joint. This approach minimized seal diameter (area) thus reducing pressure loads, which directly affect criteria of size, weight, and torque. Included in the requirements are parameters of pipe I.D., flex angle, and torque. Pipe diameter and flex angle establish a minimum spherical arc length to accommodate flexure of the primary seal, backup seal, and bearing zone. Torque is a strong function of bearing contact angle (cosine wedge angle). Radial force, resulting from axial force, is minimized with a low contact angle, however the length of the joint increases with these lower angles. The relationship of these parameters, identified on Figure 10, are derived in the equation discussed in Section 2.4, Analytical Design of a MFJ.

Other trends of design features such as separation of bearing and seals were evaluated in this trade study as well as space for stops and attachment joints. The five alternate design concepts were developed utilizing these trends as guidelines to an optimum MFJ configuration. Required flex angle was effectively reduced by half in some design concepts by independently flexing both ends of the joint. Another size reduction technique was accomplished by co-locating the bearing zone concentric with the seals (Concept #5) thus sharing the same arc length. See Figure 10 for a summation of the results of the MFJ Trade Study.

3.0 MECHANICAL FLEX JOINT LITERATURE / HISTORICAL SEARCH

The Mechanical Flex Joint design task will advance the state-of-the-art in flexible joints for cryogenic propellant feedlines. Before any effort was put forth on designing a flex joint an extensive literary search was conducted. The literary search was conducted to see if any work had ever been done on this type of design; and, if not, why. Our search helped steer the design away from known problem areas in the field of flex joints and also helped us to gain insight into other areas. Due to the massive amount of data, the literature is not included in this report, but rather included as a reference listing of those publications reviewed in the literary search. The literary search was begun by first looking at the flex joints to be replaced on the Low Pressure Fuel Duct.

3.1 SSME LOW PRESSURE FUEL DUCT CONFIGURATION

Rockwell International Corporation

The following part numbers make up the Low Pressure Fuel Duct Assembly on the Space Shuttle Main Engine (SSME). The MFJ was designed to replace the short flex joint (P/N RS008981) configuration on this assembly. Before Sverdrup began the design of the MFJ, the following drawings were carefully reviewed to establish the basis for the MFJ design.

P/N RS007018 Duct, Discharge, Insulated, LP Fuel Turbopump, Assy. of
Release Date: 6/4/81

P/N RS008981 Flex Joint, Short-Discharge Duct Fuel Pump, Assy. of
Release Date: 5/4/73

P/N RS008961 Flex Joint, Long - Discharge Duct, Fuel Pump, Assy. of
Release Date: 5/4/73

P/N RS008991 Bellows Jacket, Short, Fuel Pump Discharge, Assy. of
Release Date: 8/22/72

P/N RS008887 Bellows, Short, Fuel Pump Discharge, Assy. of
Release Date: 8/21/72

P/N RS008886 Bellows, Long, Fuel Pump Discharge, Assy. of
Release Date: 6/1/83

P/N RS008971 Bellows Jacket, Long, Fuel Pump Discharge, Assy. of
Release Date: 8/22/72

P/N RS008857 Seal, Pressure Assisted-Cryogenic
Release Date: 9/12/73

3.2 LITERARY/HISTORICAL SEARCH OF FLEX JOINT DESIGNS

A literary / historical search has been conducted at Marshall Space Flight Centers Redstone Scientific Information Center (RSIC) in an effort to locate some of the past designs of Cryogenic Bellows and Flex Joints.

The search was conducted through the massive RSIC database by using the following key words to search for related topics: CRYOGENIC, JOINTS, SEALS, FEEDLINES, FLEXIBLE, BALL JOINTS, MECHANICAL JOINTS, GIMBAL. The following are papers, articles and periodicals written on flex-joints. The number underlined is the RSIC's identification search number for the work cited.

75A45579

Issue 7 Page 1148 Unclassified

Some Unusual Oscillating Bearing Applications in Liquid Propellant Rocket Engines

80A22679

Issue 23 Page 3420 CNT# NAS8-25156 Unclassified

Nonmetallic Materials and Composites at Low Temperatures (sponsored by the International Cryogenic Materials Conference Board.)

73N22423

Issue 13 Page 1533 CNT# NAS3-12004 Unclassified

Development of Welded Metal Bellows Having Minimum Effective Diameter Change for Cryogenic Turbomachinery Face Seal Applications.

75N30245

Issue 21 Page 2636 CNT# NAS*-17796 Unclassified

Lightweight Thermally Efficient Composite Feedlines for the Space Tug Cryogenic Propulsion System

74N27369

Issue 16 Page 1972 CNT# NAS3-17796 Unclassified

Lightweight Thermally Efficient Composite Feedlines, Preliminary Design and Evaluation - for Space Tug Propulsion System.

GE 7-102-FR, Volume 1 and 2 Final Report

General Electric Advance Technology Laboratories - Proprietary
Study of Dynamic and Static Seals for Liquid Rocket Engines

Rockwell Space Division Report SD-78-AAP-01214

Development of a Cryogenic Rotating Heat Pipe Joint
J. P. Wright, Contract NAS2-9726, September 1987

Rockwell Space Division Report TSD/ZBF/85-033

Experiment Evaluation of Rotatable Cryogenic Joints
Z. F. Backovsky, 1985

Rockwell Space Division Report SPE-SEW-2000-89-030

Rotatable Cryogenic Joint Development
H. T. Nguyen and D. E. Wilson, September 1989

American Institute of Aerospace and Astronautics (AIAA) 85-0068
Thermal Design of Rotatable Cryogenic Heat Pipe Joints
E. P. French, January 1985

AIAA 90-1688
Measurement of Torque and Heat Transfer Across a Mechanical Contact Rotating Flat Surface
H. T. Nguyen, D. E. Wilson and A. Holmberg

AIAA 92-2882
Development of Cryogenic Rotatable Heat Transfer Joints
J. A. Sadunas, Z. F. Backovsky and D. E. Wilson, July 1992

Sperry Aerospace and Marine Group, Document No. 5200-021361
Rotary Heat Transfer Devices for Cryogenic Vacuum Applications
P. Jacobson

NASA Technical Memorandum 103808
Evaluation of Candidate Alloys for Construction of Metal Flex Hoses in the STS Launch Environment

3.3 COMMERCIAL VENDORS OF FLEX JOINTS

Advanced Thermal Systems - Francis H. Hulme
15 Enterprise Drive
Lancaster, NY 14086
(716) 681-1800

Advanced Thermal Systems manufactures high pressure/low temp ball joints used in industrial applications, i.e. steam line distribution and storage tank connections. Although these ball joints could meet the MFJ pressure and temperature design requirements, their excess weight does not make them worthy of flight hardware.

Arrowhead Products - Beman F. Weathers
835 Maloy Road
Williamson, GA 30292
(404) 525-0038

Arrowhead Products *Lessons Learned Design/Specifications of Flex Joints* is provided at the end of this section. These specifications were compiled at the Sept. 27, 1991 NLS Propulsion System Ducting Symposium

Allied-Signal Aerospace Company - Donald Atchison
Fluid Systems Division
1300 W. Warner Rd. - P.O. Box 22200
Tempe, Arizona 85282
(602) 893-4830

Fairchild Controls Systems - Mike Petrozzi
1800 Rosecrans Avenue
Manhattan Beach, California 90266-3797
(213) 643-4761

* *PROPRIETARY* * P/N 891099004 (Prepared for the Pressure Fed Booster)

Fairchild supplied several more drawings on related high pressure/high temperature applications. Although these parts aren't actually Mechanical Joints, Ball Joints or Bellows they have helped because they are used at similar pressures and temperatures. All part numbers are Proprietary.

P/N 74366001 Voyager Disconnect
P/N 76300001 Orbital Service Disconnect
P/N 75377001 High Pressure Helium Coupling
P/N 74338001 LOX/LH2 Overboard Bleed Disconnect
P/N 74328001 Propellant Fill and Drain Valve

Flexonics - Roger Amidon
2840 Bettis Court
Marietta GA 30066
(404) 255-7510

Ketema
790 Greenfield Drive
P.O. Box 666
El Cajon, CA 92022
(619) 442-3451

Applicable flex joints designed for the Space Shuttle/Centaur programs. All of the Part Numbers listed below have completed qualification testing.

P/N 8-031286 4 in. Gimbal Assembly

Medium: LH2

Temperature: -423 degrees F

Angulation: +/- 13 degrees

Operating Pressure: 105 psig

Release Date: 5/24/85

P/N 8-031397 5.5 in. Gimbal Assembly

Medium: LH2, LO2

Temperature: -423 degrees F

Angulation: +/- 13 degrees

Operating Pressure: 100 psig

Release Date: 3/16/84

P/N 8-050094 8 in. Gimbal Assembly

Medium: LH2, LO2

Temperature: -423 degrees F

Angulation: +/- 13 degrees

Operating Pressure: LH2 105 psig

LO2 275 psig

Release Date: 4/10/89

Litton Fastening System
3969 Paramount Blvd.
Lakewood, CA 90712
(213) 421-3711

P/N 143775 Swivel Assembly -24 Self Aligning
Release Date: 8/2/78

P/N 143780 Swivel Assembly -16 Self Aligning
Release Date: 8/3/78

Martin Marietta Aerospace
Michoud Operations

Sverdrup investigated the Vent/Relief Valve for the Space Shuttle External Tank. This valve has a seal that is exposed to an environment similar to the MFJ. The investigation determined that this seal would be of little use to the MFJ design application. The valve opens one cycle and is not considered a dynamic seal.

Pathway
P.O. Box 1526
El Cajon, CA 92022
(714) 440-1300

This company is much like other companies that make bellows type joints in the regard that they are made for commercial applications rather than man rated flight hardware. Like Advanced Thermal Systems, Pathway could probably meet our pressure and temperature requirements but the weight of the joints is a limiting factor.

Securamax - Mike Crim
333 N. Sam Houston Pkwy. E.
Suite 1150
Houston, Texas 77060
The Lippert Co. - Terry Lippert
210 Hembree Rd.
Roswell Ga. 30075
(404) 996-8710

The Lippert Company is the regional distributor for Securamax products, and all interfacing was done through Terry Lippert. Securamax's engineers did a preliminary design to meet the MFJ requirements and submitted it to Sverdrup for inspection (see Figure 8). The design was very similar to Sverdrup Concept 5 and is a very unique design. Below is some information on this drawing for future reference.

P/N p15-109 Securamax Flexi-Ball proposal
NASA MSFC Reference Number: SFQ 91-0523
Release Date: 05/23/91

The typical Flex-Ball joint is a commercial ball joint that has 30 degrees of total axial pivot with 360 degrees of rotation. The Flex-Ball can withstand pressures over 2000 psi at 650 degrees Fahrenheit, but the crushed graphite rings used for seals could pose a problem at the low temperatures the MFJ would require.

Stainless Steels Products - Dick Sigerist
2980 No. San Fernando Blvd
Burbank, CA 91504

P/N ME271-0012-0009

Supplier P/N 1803735-102

External Ball and Socket for Saturn V S-II Stage Line Assembly, Fill and Drain, Liquid Oxygen

P/N 1007631 Universal Rotational Joint Assy 5.00 in. Diameter

Release Date: 7/26/73

This design is very close to meeting the MFJ design requirements. It is included in the trade study (see Figure 9).

P/N 1007631 URJ B1-B 5 in. Diameter Warm Air Duct

3.4 PERSONAL EXPERIENCE DESIGNING BELLOWS

The following section is a record of personal contacts Sverdrup has had with engineers experienced in the area of bellows and flex joints. The following notes are not official but rather notes made during telephone conversations and personal contact. They are included in this report as reference to scientific information and should not be copyrighted.

Stainless Steel Products

Art Moore

-Stainless Steel Products published MFJ in 1976 regarding Ron Urquidi and Chuck Daniels of Rocketdyne

-Mr. Moore does not know of MFJ publication.

-Mr. Moore does not know of any MFJ cryogenic application without bellows as a seal.

-Mr. Moore will discuss our requirement in new products meeting.

-Sent 12 Jan. 76 article from Design News. Universal Rotational Joint (URJ) 1 million motion cycles. 20 degrees/360 rotational B-1 engine bleed air duct @ 650 degrees F. Used spring-loaded Fluoroloy "K" jacket seal.

Iraj Maroofian, Engineer

-Has designed 8" / 140 psi cryogenic joint that weighs 20 lb. Feel certain they can build the joint we want and to our specifications. Can not release this particular drawing because it proprietary.

D. Thompson - Marketing

-Sent Stainless Steel Products brochure and technical bulletin "Low Bending Moment Ball Socket Joint." This particular joint has a bellows type seal. Bulletin contains torque curves of ball / socket joints.

Chemical Propulsion Information Agency (CPIA)

Lee Viper

-Mr. Viper searched Engine Design and Fuel Systems.

-Found no reports on cryogenic MFJ's.

-Found report August 1985 to August 1987 Rockwell (Downey) F04611-85-C-0052/AFAL TR-87-055, "Lightweight Propellant Feed System," which concluded "there is no alternative to metal bellows."

Rocketdyne

Ron Urquidi - Recommended the following approaches regarding seals for MFJ:

- spring loaded
- little pressure assist
- Teflon/hybrid material
- omni coil by Fluorocarbon (renamed to Furon)
- threaded coupling (axial alignment)
- Vetco-Gray "T" seal flange w/out bolts
- acceptable leakage rate (collect/discharge)

Don Stuck - Seals Expert

- no history of movable cryogenic seal
- NAFLEX not used as movable seal
- metal movable seal need lubricant (silver or dry film not recommended)

Chuck Daniels, Ret.-MFJ Designer

- Experience closest is lip seal on quad valve. This is the Vent/Relief Valve manufactured by Martin Marietta (See Commercial Vendors of Flex Joints)
- Danger of high leak rate if Teflon seal should crack in high pressure application.
- U-seal similar to Bal Seal
- Bad experience with spiral seal in static application under dynamic loading. The spiral will spin out of seal jacket material. These type of seals are not used on the Shuttle, NASP, ALS
- Crush seals, even static, have no resiliency and will not pass signature test after firing

Miles Butner - Bearing Expert

- Said to consult Fred Doland @ MSFC Materials Lab
- Breakaway torque under high bearing load of ball on flat plate
- Coefficient of Friction:
0.0015 ball w/o rubbing or yielding
0.5 two balls rubbing (opposite)
0.2 non-metal or bronze retainer

Vetco Gray Inc.

Frank Adamek - Program Manager/Aerospace Division (Houston)

- No movable cryogenic experience/record.
- Static cryogenic experience:
Stennis
Shock tube (liquid Helium) 22 inch
MBB, Germany has 8 foot U-seal .06 in. axial travel
Recommended dual ion beam coating for low friction.

Securamax International

Mike Crim - Vice President of Sales

- Capabilities
Design
Finite element analysis
CNC manufacture
Testing
Valve Ball ratio = 0.6 I.D./O.D.
Has aerospace experience and Can-Do attitude
- Has not made cryogenic flex joint
- Would approach like a ball valve

- Finish on ball valve surface is 2 micro-inch
- Requested weight limit

The following are lessons learned from vendor Arrowhead Products.

Arrowhead Products

NLS Propulsion System Ducting Symposium Sept. 27 1991

Lessons Learned Design/Specifications of Flex Joints:

Pre-RFP supplier specification content involvement

Design for producibility

Avoid Fillet Welds - Low stress - No X-ray

Avoid dissimilar metals

Avoid Vacuum jackets

Avoid flow liners for cleanliness

Avoid Austenitic Stainless Steel Bellows

Avoid unnecessary stringent specification requirements:

i.e. Flange sealing features

X-ray discontinuity Allowable

Lessons Learned Manufacturing/Processes

Write Material and process specifications geared specifically toward product and facilities

Emphasize contamination control

Apply Statistical Process Control (SPC)

Maximize use of Automated Fusion Welding

RECOMMENDATION: Conduct a literary search to insure some valuable information on the MFJ is not being overlooked.

4.0 SEALS:

The ability to seal the MFJ is one of the major obstacles in this design effort. The following section is dedicated to seals and some of the variables the designer must take into consideration.

4.1 DESIGN VARIABLES FOR SEALING

Seal Material Permeability - Permeability is the tendency of gas to pass or diffuse through the elastomer. The permeability of the seal jacket material being used is expressed in values of $P \times 10^{-10} \text{ cm}^3/\text{mm}/\text{cm}^2/\text{sec.}$ or cubic centimeters of gas under normal conditions traversing 1 cm^2 of surface, 1 mm thick, per sec., per cm of mercury pressure. This is an important aspect in determining if a seal will be appropriate for the job to be accomplished. After determining this acceptable leakage rate, the permeability rate of various seal jacket materials must be determined. If the permeability rate of the seal jacket is greater than the acceptable leak rate, the necessary adjustments must be made. These adjustments will vary from aborting the test to finding new seal jacket materials and adjusting gaps, surface finishes, and hardnesses.

Some other variables that should be taken into consideration are the mechanical properties of the seal jacket material. Listed below are some design and seal elastomer variables and a brief definition that should be studied extensively when designing for cryogenic temperatures.

Resilience - ability of the material to return to its original state or maintain contact pressure during service.

Elongation - increase in length expressed numerically as a percent of initial length. This property primarily determines the stretch and the ability to return to the original state.

Modulus of Elasticity - stress at a predetermined elongation, usually 100%. It is expressed in pounds per square inch.

Tear Resistance - resistance to cuts or ruptures of seals. Seals with poor tear resistance will fail more quickly under further flexing or stress, once a crack is started. Inferior tear strength of a seal jacket is also indicative of poor abrasion resistance.

Abrasion Resistance - The wear resistance of a compound related to scraping or rubbing of the surface. This surface does not necessarily have to be the sealing surface but can be a surface the seal contacts during installation. Special care should be taken to make sure there are no sharp corners in the area of seal installation. This variable is especially critical when bearing and sealing surfaces are shared.

Coefficient of Friction - Coefficient of friction of a moving rubber seal relates to hardness, lubrication and surface characteristics of surrounding materials. This is a critical design item when selecting a seal jacket material for cryogenic environments. Breakaway friction should be taken into consideration.

Coefficient of Thermal Expansion - Coefficient of thermal expansion is the ratio of the change in length per degree F to the length at 0 degrees F. Typically elastomers have a coefficient of expansion ten times that of steel. This is another critical issue when designing with dissimilar materials, i.e. elastomers and Inconel or stainless steel.

Shaft Coatings - Chrome is not recommended but if used, it should be hard, nonporous, and very smooth. Chrome should be used in light duty conditions only. Electrolyses nickel should be used in the as-deposited condition only. Plating on hard substrates is preferred over plating on soft materials.

4.2 DSTF VENDORS

The Dynamic Seals Test Fixture (DSTF) requires four different face seals, two of the seals are dynamic and the other two are for a static application. Listed below is the information sent to the seal vendors in order to receive their bids.

*Reference Section 5.2 for information on the DSTF and how it works.

Seal Sizes: Static; 11 in. Dia. & 8.5 in. Dia. / Dynamic; (2)7.5 in. Dia. & (2)1.5 in. Dia.

Design Temperature: -415 degrees F

Maximum Design Pressure: 343 psia

Maximum Operating Pressure: 326 psia

Service: Reciprocating / Static

Cycle Life: 400 full angular cycles @ 343 psia and -415 degrees F

2800 non-operational @ ambient temperature

Seal material compatibility: Liquid Oxygen (-320 degrees F)

Liquid Hydrogen (-423 degrees F)

Helium (-452 degrees F)

Cycle stroke: 2.00 inches

Cycle rate: 1.22 cycles/second

External Leakage: Shall not exceed 1×10^{-3} scc/sec GHe when pressurized to the design pressure with Helium for temperatures ranging from -420 degrees F to 140 degrees F.

Recommended surface finish: 2 Ra

Recommended gap/clearance between piston and sleeve: .002 in.

Other variables: Surface hardness Rockwell C

Seal material permeability

Section 4.3.2 contains the list of the seal prices submitted by seal vendors for the DSTF. The original list of vendors was selected from the Thomas Register. After consulting with several engineers from Sverdrup with extensive aerospace experience, the list has been narrowed down to the three vendors listed. Included is the part number, diametrical dimension of the seals, and the price. The vendors recommended surface finish and gap are also included. The prices on these quotes have expired.

4.3 SEAL EVALUATION DATA

4.3.1 Seal Vendor Specifications:

Seal Vendor	Recommended Surface Finish	Recommended Gap(in.)
<i>Furon Seal</i>	4 - 8 RMS	.002
<i>American Vari-Seal</i>	4 - 8 RMS	.002
<i>Bal Seal</i>	2 - 4 RMS (1.8 - 3.6 Ra)	.012 @ 70 degrees F

4.3.2 Seal Vendor Prices (1993):

Seal Vendor	1.5 in. Seal	7.5 in. Seal	8.5 in. Seal*	11 in. Seal
<i>Furon</i>	\$61.49	\$231.91	\$144.94	\$256.69
<i>American Vari-Seal</i>	123.86	616.95	777.03	960.42
<i>Bal Seal</i> (FEP)	123.20	406.60	532.05	721.35
(UPC)	11.60	203.30	228.75	328.95

* 8.5 in. Seal should be replaced after each test of 7.5 in. Seal

4.3.3 Seal Vendor Materials:

Furon: Proprietary Fiber Glass Filled (PTFE)

American Vari-Seal: KEL-F 81 (PCTFE)

Bal Seal: Fluorinated Ethylene Propylene (FEP)
Polyethylene (UPC)

4.3.4 More Specifics of Seal Vendors:

Furon Seals

4412 Corporate Center Drive

P.O. Box 520

Los Alamos, CA 90720

(714) 995-1818

6 Weeks Delivery

P/N 230108186	1.5"	\$61.49
---------------	------	---------

P/N 230108185	7.5"	\$231.91
---------------	------	----------

P/N 380002877	8.5"	\$144.94
---------------	------	----------

P/N 3880002876	11.00"	\$256.69
----------------	--------	----------

Suggested reading from Fluorocarbon Mechanical Seal Division
Omni Design Handbook

American Variseal Corp. - Shamban - Go Air

510 Burbank Street

P.O. Box 1479

Broomfield, CO. 80020

4 - 5 Weeks Delivery (Min order of 4)

Recommended their dual cantilever spring with Turcite 37 seal jacket.

P/N S67200-1177	7.5"	\$616.95 ea.
-----------------	------	--------------

P/N S67200-1178	1.5"	\$123.86 ea.
-----------------	------	--------------

P/N IF-D-08500-37S	8.5"	\$777.03 ea.
--------------------	------	--------------

P/N IF-D-11000-37S	11.00"	\$960.42 ea.
--------------------	--------	--------------

Ball Seal Engineering Company

620 West Warner Avenue

Santa Ana, CA 92707-3398

1 - 5 Weeks Delivery (Min. order of 4)

Permeability: FEP=30.1 X 10⁻⁸ @ 77 degrees

UPC= unknown

P/N X36672	1.5"	\$123.20 ea.
------------	------	--------------

P/N S17HBA-(8.00)-FEP	8.5"	\$532.05 ea.
-----------------------	------	--------------

P/N S17HBA-(11.00)-FEP	11.00"	\$721.35 ea.
------------------------	--------	--------------

P/N S17HBA-441-FEP	7.50"	\$406.60 ea.
--------------------	-------	--------------

P/N 314MB-128-UPC	1.5"	\$11.60ea.
-------------------	------	------------

P/N S17HBA-(8.00)-UPC	8.5"	\$228.75 ea.
-----------------------	------	--------------

P/N S17HBA-(11.00)-UPC	11.00"	\$328.95 ea.
------------------------	--------	--------------

P/N S17HBA-441-UPC	7.500"	\$203.30 ea.
--------------------	--------	--------------

Suggested reading from Bal Seal:

Technical Report #75 Shaft and Housing Materials, Coatings, and Lubricants for
Cryogenic Service

Technical Report #74 Sealing at Cryogenic Temperatures with Bal Seals

Technical Report #78 An Analysis of the Factors Which Influence Seal Performance

Technical Report #31 Optional Quality Assurance Procedures and Documentation

Technical Report #4A The Influence of Surface Finish on Bal Seal Performance

Catalog No. 1.8 Cryogenic Seals
Catalog No. 5.1A Static Face Seals

RECOMMENDATION: Investigate other aerospace seal vendors with experience in cryogenic temperature dynamic face seals.

5.0 DYNAMIC SEALS TEST FIXTURE

As mentioned earlier, the problem of sealing the MFJ is one of the biggest obstacles to a successful design. The seals will have minimal experience in our design environment and redundant seals will possibly prevent a single point failure of the MFJ. Since the issue of seals is a "gray area" it has been decided to test the seals before an expensive MFJ is designed and built. Sverdrup's approach includes the design of a very simple cylindrical test fixture to test seals similar to the ones to be used in the MFJ.

The cylindrical test fixture is designed to use a similar seal as the MFJ Concept 5, which has a seal diameter of 7.5 inch. It is worth noting that the MFJ will require a seal to "seat" on a ball and socket configuration, while the Dynamic Seals Test Fixture (DSTF) seals will "seat" on a cylindrical surface. The trade off between these two is that a "Ball and Socket" configuration would be very expensive to build, and would have less predictable geometry and clearance gap. Although the DSTF will not emulate the exact concentricity of MFJ Concept 5, it will be exposed to the same harsh environment, and will provide an adequate test to demonstrate whether the seals will be acceptable. If the recommended selected face seals failed at this point, alternative seals will have to be evaluated or development of the MFJ stopped.

After extensive investigation it was discovered that there had never been a dynamic seal used at the temperatures and pressure the MFJ would demand. Several seal vendors were very interested in taking on the challenge of designing a seal that "would work." Basically the seal vendors were going to try to use their static seals in a dynamic environment. The way this would be accomplished is to tweak some of the known variables, i.e. surface finish and gap, in an attempt to make these static seals work. The short duty cycle is favorable and permits high contact spring force. See Section 4.0, Seals, for specifics on the seals to be used and the vendors.

The following section has been compiled to help designers review what has been learned while designing the DSTF. The following tips are abbreviated and further investigation of each of them is recommended. The first thing listed is a review of the MFJ design issues and how they apply to the DSTF. Reference these design requirements in Section 2.1.

The original plan was to design the DSTF to have the ability to test the same 7.5 in. face seal under a variety of conditions. The conditions that were going to be changed are the gap and surface finish of the test sleeve (P/N 96M66993-15). These are two variables the designer has control over, and by taking best and worst case conditions, it is helpful to establish failure modes and/or acceptable leakage rates. The original plan has three test cylinders ranging from best case (smallest gap and smoother surface finish) to worst case (wider gap and rougher surface finish). The gaps have never been established because a thermal analysis has not been conducted. For more information concerning seal vendors specifications, reference Section 4.3, Seal Vendor Evaluation Data Sheet.

5.1 DSTF DESIGN ISSUES:

- a.) Movable Cryogenic Seal
- b.) Stress/Thermal Shock
- c.) Dynamic Loading
- d.) Fabrication
- e.) Reliability
- f.) Stress Analysis
- g.) Insulation / Movable Insulation
- h.) Leak Detection

- i.) Non-Hazardous Test Medium
- j.) Low Temperature Test Limit.

The above design issues are critical when designing a MFJ; therefore, they are critical for designing the DSTF. By addressing these issues when building the DSTF, it is believed it will help when building the MFJ. Although each of these variables is critical to designing the MFJ, all of them have not yet been addressed.

5.2 HOW DOES THE DSTF WORK

As mentioned above, this test fixture is a simple cylindrical seals test fixture. To simplify the explanation of how this test fixture works, a review of P/N 96M66993 drawing package in Appendix D. After reviewing this package, one should consult the cross-section on sheet #1 of P/N 96M66993-01. The test seals are placed on the piston configuration (P/N 96M66993-29) and moved in an up-and-down motion. There is two 0.375 in. vertical drill holes in the piston to allow the gas to pass between the upper and lower part of the piston as it moves up and down within the cylinder. The 0.25 in. diameter hole drilled the entire length of the piston is used to pressurize (343 psia) the seal cavity from an outside pressure source. During a test the pressure source will have a set pressure of gaseous helium. The helium will be measured for leak rate with a mass spectrometer and will be used to determine if there is leak in the test seals. After understanding where the test seals are located, how the piston moves and how the seals are pressurized, the other features will now be reviewed. The pistons seals interface with the seal test sleeve (P/N 96M66993-15), which has a 2 micro-inch surface finish. To obtain the extremely low temperatures, the MFJ will be exposed to LN2 or LH2 to chill down the area of the seals. Part Numbers 96M66993-19, 96M66993-17, 96M66993-23 and 96M66993- 21 make up the containment area for the cryo-bath to chill the seals to a test temperature to -320 degrees Fahrenheit (LN2) or to -415 degrees Fahrenheit (LH2). Reference P/N 96M66996-01 drawing for the entire assembly and its design features.

5.3 DSTF MATERIAL SELECTION

The DSTF is designed of materials to emulate the MFJ. The MFJ, like the current bellows configuration, should be constructed of the same material as the Space Shuttle Main Engine Low Pressure Fuel Duct. Currently the duct is constructed of 21-6-6 CRES but may be changed to Inconel 625. Therefore, one of these will be the material with which the DSTF will be constructed. After investigating the price of the Inconel 625, it is recommended a feasible, study be conducted before constructing the entire seal test fixture from this material. In an effort to make the price of the test fixture feasible, a search for alternative materials has been conducted. After much investigation it has been decided to make the DSTF piston (P/N 96M66993-29) and inner sleeve (P/N 96M66993-15) out of Inconel 625. The remaining DSTF is to be made of 304L SS. 304L was chosen because it was readily available and low cost, and because the low carbon content makes it very desirable for welding. The biggest concern about making the test fixture out of Inconel and 304L is the coefficients of thermal expansion between dissimilar metals. The difference in thermal expansion of these materials will affect the gap between the cylinder sleeve (P/N 96M66993-15) and sleeve support (P/N 96M66993-23). In order to obtain a snug fit at LN2 temperature of -320 degrees F the ambient temperature gap must be about 0.0047" (diametrical).

5.4 DESIGN CONSIDERATIONS FOR DYNAMIC CRYOGENIC SEALS

There are several considerations to take into account when designing hardware to be used in a cryogenic sealing application. Although there are many variables to consider, surface finish, test fixture materials and face seal spring pressure are the three most critical on this particular task. The reason these are critical on this job is that they are three variables that the designer can control. Although permeability, elongation, and other variables are major factors in the success of the test, there is limited control due to material limits for cryogenic applications. The next two sections review some variables the designer can control.

Surface finish - The surface finish of hardware in contact with the seal will have a direct effect on sealing performance. Smooth finishes provide better sealability, less wear, and lower friction. Generally, dynamic surface finishes should be two times as smooth as static surface finishes, i.e. 32 micro-inches Ra for static surface and 16 micro-inches Ra for dynamic surfaces. For high speed applications a surface finish of 8 micro-inches Ra is normally recommended. Where lubrication is present, finishes less than 4 micro inches should be avoided. The MFJ and DSTF are considered to have a low cycle rate when testing cryogenic and low molecular gases and therefore, the three seal vendors recommended a 2 - 8 micro-inch surface finish. For more specifics, see Section 4.3 on Seal Evaluations Data. The cost of machining these types of surface finishes will be a major contributor to the cost of the DSTF and should be further investigated when procurement is necessary.

Clarification of Surface Finish: Micron and micro-inch are not interchangeable terms.

Micron - 1/1,000,000 of a meter or 0.000039 inches (39×10^{-6}) inch

Micro-inch - 1/1,000,000 of a inch or 0.000001 inches (1×10^{-6}) inch

Clarification of Measuring Surface Finish: Ra and RMS are not interchangeable terms.

Both of these methods measure surface finish in units of micro-inch but each use a different method to determine the roughness.

Ra (Roughness Average)

RMS (Root Mean Square)

Suggested Reading on Surface Finish:

1.) *ANSI B46.1 1985 Surface Texture* (Surface Roughness, Waviness, and Lay)

2.) *Rod Surface Profile and Coating Effects on Seal Performance*

Report No. R1068, November 23, 1992, Shamban Seal Division

3.) *Surface Finish Technology for Hydraulic Seals and Actuators*

1992 - International Fluid Power Exposition

RECOMMENDATION: Extensively study surface finishes and their effects on dynamic cryogenic seals.

Surface Hardness - Mating surface hardness directly affects the seal life, especially in high speed applications. Since high speed will not be a variable in designing our hardware, surface hardness will not be a major consideration in our design.

Shaft Coatings - Chrome is not recommended, but if used it should be hard, nonporous, and very smooth and used in light duty conditions only. Use electrolyses nickel in the as-deposited condition only. Plating on hard substrates is preferred over plating on soft materials.

Dissimilar Metals - See Thermal Analysis, Section 5.5.2.

The DSTF has a PDR quality set of completed drawings and have been presented to NASA's MSFC. Shortly after presenting this drawing package, MSFC personnel began the process of a stress analysis when the project was terminated. A complete listing of the drawings that comprised the DSTF is listed in Section 5.7.

5.5 STRUCTURAL ANALYSIS

Sverdrup has recommended the following structural analysis be conducted on the DSTF. Below is a brief synopsis of each analysis.

5.5.1 Stress Analysis

As mentioned above, Sverdrup presented NASA MSFC personnel with the PDR set of drawings, which were in the process of getting the stress analysis begun when the project was terminated. Although a formal stress analysis was not performed, there were some preliminary calculations done on the DSTF. After these calculations were conducted it was determined that the inner bottom plate (P/N 96M66993-07) and plunger guide (P/N 96M66993-24) thickness should be changed from 0.375 inches to 0.875 inches. Preliminary calculations indicated the 0.375 inch plate is sufficient, but Sverdrup's Structures and Dynamics personnel believed with the given test pressures, the plates should be thicker. This presented a problem because a thicker plate would cause problems with thermal gradients when chilling down the test article. In an effort to compromise between the plate thicknesses, pressure, and thermal gradients, we opted to put a pressure relief valve on the plunger guide. The pressure relief valve would be located 180 degrees from the mass spectrometer interface and would be opened at a predetermined set pressure. After the plates thickness was increased, it was also suggested to change the method that the inner bottom plate (96M66993-24) and the inner wall (96M66993-23) were welded together. The suggested method of welding can be found in the ASME Boiler Code Handbook on page 115, Figure C or D. The calculations are preliminary and parts should not be changed until more definitive calculations have been conducted. RECOMMENDATION: Conduct a formal stress analysis before a CDR package is produced.

5.5.2 Thermal Analysis

Several variables must be taken into consideration when designing the DSTF. Due to lack of funding, this test fixture has not received a thermal analysis. The following is a list of things the thermal analysis should take into consideration:

1.) As mentioned earlier, the test fixture should be constructed of dissimilar metals. The difference in coefficients of thermal expansion between Inconel 625 and 304L is one of the biggest concerns from a thermal perspective. The materials thermal expansion will affect the gap between the piston (P/N 96M66993-29) and the test sleeve (P/N 96M66993-15). If 304L cylinder (P/N 96M66993-23) contracts faster than the Inconel test sleeve (P/N 96M66993-15), severe deformation of the entire test article could occur.

RECOMMENDATION: Consult MSFC-SPEC-256 and MIL-STD-889 on the relationships between Dissimilar Metals.

2.) Another variable is the amount of chill time necessary to chill the inner piston to the temperature that emulates the current flex joint environment. This is critical for two reasons: First, so that the proper temperature is reached for testing the seals; and secondly, so that the test sleeve and the piston are within a desired temperature range before activating the test. The later reason is critical because the test sleeve will chill down first and actually contract around the piston. If the piston does not have time to chill down and the piston begins to move the piston could mar or scratch the test sleeve or damage the piston.

3.) How much insulation is going to be needed on the Dynamic Seals Test Fixture? The amount of insulation will directly effect the amount of thermal losses because of its effect on heat dissipation from the Dynamic Seals Test Fixture. Another issue, other than the amount and type of insulation, is the problem of the movable insulation that will be

necessary where the piston shaft enters the plunger. For more information on the insulation, see Section 7 on insulation.

RECOMMENDATION: Conduct a formal thermal analysis before a CDR package is produced.

5.5.3 Weld Fatigue Analysis

A weld fatigue analysis should be conducted on the DSTF to insure structural integrity. One area of possible concern is where the standoff tabs (P/N 96M66993-28) is welded to the bottom plate (P/N 96M66993-17). The up-and-down motion of the actuator could cause irregular deflections and possibly cause the welds to fail.

RECOMMENDATION: Conduct a weld fatigue analysis before a CDR package is produced.

5.6 CHANGES IN THE ORIGINAL DYNAMIC TEST SEAL STRATEGY

The original test plan included several things that have since been changed for various reasons. These applications may be of use in another design effort.

5.6.1 Controlled Environment

As mentioned in the Thermal Section of this report, it will be very difficult to determine whether the piston (P/N 96M66993-29) has chilled to the correct temperature.

5.6.1.1 Heaters

The original design included heaters to be attached to the inner wall (P/N 96M66993-23). If the worst case seals test fails, these heaters would be used to "warm" the test fixture until the seals began to work. The two thermocouples on the outer wall of the test fixture are used to determine the temperature within the cryo-bath.

After reviewing the original scope, it was decided that if the seals did not perform at the operational temperature of the MFJ, they had failed. With that in mind, the decision was made, to reach the desired temperature and not to continuously regulate the environment.

Since the heaters are no longer a part of the DSTF, it has also been determined that the bottom plate (P/N 96M66993-17) could be welded to the outer cylinder (P/N 96M66993-19). Welding these two parts together eliminates another leak path or failure point and also eliminates the need for the 11.5 in. static seal. After reviewing this design, the decision was made to leave the current configuration to allow for possible future maintenance of the test fixture.

RECOMMENDATION: In an effort to eliminate another failure point, weld P/N 96M66993-19 and P/N 96M66993-17 together. This is only recommended if heaters are never planned to be used.

5.6.1.2 Single vs Multiple Test Sleeves

Originally three different test sleeves were to be incorporated into the test configuration. These three differing test sleeves were to have varying gaps and surface finishes, i.e. smallest gap and smoothest surface finish (2 Ra), as well as a wider gap and rougher surface finish (8 Ra). The decision was made to try the "best case" sleeve (smallest gap and smoothest surface finish) to see if it would work. This single test would give us the desired information. If the decision is made to try another gap/finish configuration, the test sleeve could then be built. This additional cost is another reason the decision was made not to build multiple test sleeves.

NASA Cost and Procurement group estimated it would cost approximately \$11,000 to get one test sleeve built with a 2 micro-inch finish. This price estimate included \$1371.45 for the Inconel raw material and 320 hours of machining at a cost of \$30.00/hr. The cost of

additional test sleeves with "rougher" surface finishes would obvious take less machine time but would still be a significant cost to this program.

When combining the issues of heaters, numerous test sleeves, and three different brands of seals, it was decided that too many variables were being introduced into this once-simple seals test. As the test now stands, the seal testing provides one chance to test the seals under the harshest environmental conditions and the best sealing conditions.

5.7 DSTF (PDR) DRAWINGS

The following DSTF PDR drawings are provided in Appendix D

96M66993-1 MFJ Seal Test Fixture, Sheets 1-10
 96M66994-1 Test Fixture Strong back, Sheets 1-4
 96M66995-1 Actuator Interface, Sheets 1-2
 96M66996-1 Final Assembly

5.8 PROCUREMENT OF DSTF

Estimated cost includes hours to build the DSTF includes machining, sheet metal work, welding, cleaning, assembly, and materials cost. NASA MSFC's Materials and Processing Branch (EP52) provided SvT and EP64 with two price estimates: one to be built out-of-house and one to be built in-house, or at MSFC.

<u>Part Number</u>	<u>Assy</u>	<u>Mach</u>	<u>Sheet mtl</u>	<u>Weld</u>	<u>Cleaning</u>	<u>Matrl</u>
96M66993-1	64hrs	972hrs	8hrs	36hrs	28hrs	\$6,370.66
96M66994-1		86	28		4	330.00
96M66995-1	2	34			3	80.00
96M66996-1 Assy	8					
<u>Moog Actuator (provided)(*1)</u>						
Total	74hrs	1,092hrs	36hrs	36hrs	35 hrs	\$6788.66
Total hours 1289hrs.(*2)						

Labor 1289hrs. @ \$30hr = \$45,459.00

Materials 6,789.66

Total built *out of house* \$52,247.66

Total built *in house* \$6,789.66

(*1) Moog Actuator, Model No. 17-109, Part No.9216

3000 psig

2.42" stroke

9/10 error in stroke

8.91"/sec @ no load velocity

(*2) 1289 hrs includes 16 hrs of "other" on assembly of 96M66994-1

The price to have the test fixture built in-house includes only the cost of materials. NASA MSFC has the facilities to build this test article and pay only for the price of materials.

When Sverdrup began the conceptual design of the DSTF, there was a concern of the availability of large diameter Inconel SS and 304L SS. The diameters to be referenced can be found on P/N 96M9933-19 and 96M66993-15. After referencing the Thomas Register it was discovered that A&P Alloys does provide the seamless cylinders in the diameters

desired. The 304L SS averaged \$500.00 a foot, while the Inconel 625 is \$1200.00 a foot. Consult A&P Alloys for today's market price of these materials.

A&P Alloys

400 West Street

West Bridgewater, MA 02379

1-800-221-0786

RECOMMENDATION: Solicit some outside vendors to bid on building the DSTF.

5.9 MODIFIED DYNAMIC SEALS TEST FIXTURE

A meeting was held on February 25, 1993, to present NASA EP64 with a status of the MFJ DSTF. After presenting the current status an interest was expressed in possibly modifying the current test fixture to accommodate a variety of seal sizes. The current design was modified to accommodate a variety of seal sizes ranging from 4 inches to 7.5 inches. Although the test fixture was modified, Part Numbers 96M66993-15 and 96M66993-29 would have to be rebuilt for each new diameter seal. A conceptual design drawing was presented to EP64 with the pros and cons of this design. (See Figure 11)

RECOMMENDATION: To test numerous size seals, consider this design approach.

6.0 BEARING AREA COATINGS

6.1 DIAMOND FILM PROCESS FOR MFJ APPLICATIONS

6.1.1 Summary

This section is directed to consideration of three types of thin film coatings as concept design alternatives to multiple rows of ball bearings in reducing frictional torque in a cryogenic MFJ (See Figures 3 - 7). A literature search of publications identified authors pursuing manufacture and testing of low friction coatings. A telephone survey of these authors was conducted to obtain current publications of their work, to identify similar applications, and to solicit expert opinions concerning application of their process to the baseline MFJ design. Three types of film coatings are compared: Chemical Vapor Deposition (CVD) film, Diamond-Like Carbon (DLC) film by dual Ion-Beam-Enhanced Deposition (IBED) process, and the Diamond thin film by Plasma Assisted Chemical Vapor Deposition (PACVD) process. Independent laboratory data supports manufacturers' claims of low friction coefficient with all three coatings. CVD processes have been enhanced in recent years along with related new developments of Diamond film and Diamond-Like Carbon film. Current crystalline diamond film technology development is not targeted towards bearing/low friction coatings but to larger markets such as X-Ray optical windows, cutting tools, thin high voltage insulators, high temperature electrical insulators, and heat sinks.

Molybdenum Disulfide coating applied by the proven CVD beam sputtering process will provide low friction; however, the bond may not be as good as the enhanced beam process. Diamond-Like Carbon film offers excellent mechanical film bond, low friction, and hard wear surface. Diamond film must be applied to steel over a substrate to avoid carbonization. Selected MFJ material is Inconel 625 which contains cobalt, which is not compatible with this film process. Current and near term diamond reactors are too small (eight-inch diameter) to process the MFJ baseline size. All three of these films offer low friction coefficients that are comparable to ball bearings. Film thicknesses are negligible and can be applied to a finely finished surface.

Conclusions from this survey are: DLC film applied by the IBED process is capable of providing uniform thin film with superior bond, good wear resistance, low friction coefficient, and can be applied with existing equipment.

6.1.2 Introduction

This section is a collection of data in response to an action item following Sverdrup presentation (February 1993) of results from their Mechanical Flexible Joint MFJ design trade study under Task Directive 324-005. Several design approaches were evaluated and dimensions and torque were modeled to select the optimum (MFJ) geometry. High friction is inherent with high operating pressure and large diameter movable joints.

Torque was calculated to exceed specification limit considering steel-on-steel ball joint construction. Acceptable torque can be achieved with the addition of a friction reducing bearing or low-friction film coating. Coating operating environment includes: compatibility with cryogenic fuels, resistance to severe thermal shock, high bearing strength, and bond strength at extremely low temperature. Resin bonded coatings were eliminated from consideration due to poor bond strength and large film thickness variations.

Companies engaged in development of a new low cost diamond film process (PACVD) responded to a questionnaire survey by Jassowski, Aerojet. He examines broad applications of diamond film to rocket propulsion and suggests applications to liquid fuel rocket propulsion cryogenic feed lines. Contacts with film development companies and film friction testing laboratories were identified; however, most were reluctant to offer test data considered "competition sensitive." Survey responses and bibliography were the initial source of contacts investigated for information on film applications to cryogenic feed line components.

This section presents opinions of authors following from both telephone conversations and review of their recent published works pertaining to process development and testing of low friction coatings. None of the applications was found to be similar to a cryogenic MFJ. Related/competing film processes were pursued and the three most promising candidates were selected for comparison. Design requirements of the MFJ and physical properties of candidate coatings are compared in Section 6.1.3, Film Deposition Processes.

6.1.3 Film Deposition Processes

The three thin film deposition processes considered for MFJ cryogenic bearing application are as follows: 1) Chemical Vapor phase Deposition (CVD), 2) dual Ion-Beam-Enhanced Deposition (IBED), and 3) Plasma Assisted Chemical Vapor Deposition (PACVD). Application techniques have been enhanced in these processes to permit film properties to be tailored to specific requirements. Independent laboratory tests support the manufacturers' claims of low friction coefficients.

6.1.3.1 CVD PROCESS

Beam Alloy and many other companies produce film coating with the Common Vapor Deposition (CVD) process. Direct ion implantation begins by feeding a small stream of molecular gas into an ion source assembly. Energetic electrons emitted from a filament collide with the feed gas molecules, stripping an electron off each atom to form ions. The ions are then electrostatically extracted from the ion source, formed into a beam, and accelerated to high velocities using an electrical potential. The ion beam is then directed onto the surface of the components to be implanted.

6.1.3.2 IBED PROCESS

Beam Alloy Corporation of Dublin, Ohio, produces polycrystalline diamond film and also a diamond-like carbon film. Technology and equipment exist to apply a thin film diamond-like carbon coating onto a MFJ bearing with the potential for very low friction coefficients. Development work in this new diamond film technology is aimed at other markets and little work has been directed towards bearing applications.

By comparison, ion-beam-based techniques require much more complicated and costly hardware. Deposition rates are slower, but these techniques may eventually find more applications because processing temperatures are much lower, and the process is not as sensitive to substrate composition.

6.1.3.3 PACVD PROCESS

Development of high-energy plasma assisted thin-film deposition processes (PACVD) achieved high temperature, which enabled researchers in Japan to develop the technique to

grow diamond film in 1981. (Advanced Materials and Processes, Dr. Arnold H. Duetchman, June 1989.)

"The mechanical, electrical, optical, chemical and thermal properties of diamond make it attractive in applications ranging from wear-resistant coatings for mechanical and optical components to substrates for advanced semiconductor devices."

Crystallume, Menlo Park, California, began producing polycrystalline diamond film in 1989. PACVD techniques are much easier to implement from an equipment standpoint and are, therefore, the most popular. The hardware is relatively inexpensive, and fairly high deposition rates are easily achieved. PACVD, however, requires high temperatures; and, because it relies on epitaxial film growth, the range of substrate materials that can be coated is somewhat limited.

RECOMMENDATION: Process/Equipment, Film Characteristics and MFJ Applications for the three thin film deposition processes should be further investigated.

6.1.4 Other Low Friction Films

RECOMMENDATION: Other low friction films should be investigated for the MFJ application. Teflon, ball bearings and polymer films are three candidates that merit further investigation.

NOTES ON DIAMOND FILM COATINGS:

The following notes were compiled during telephone conversations with experienced personnel in the diamond film industry. Anyone that continues work in this area should contact these persons to verify the following information.

Laurie Conner, V.P. Marketing & Sales, Crystallume, Menlo Park, California.
(415) 324-9681, 31 Jan. 92
6% cobalt tool steel, -iron is a problem
1 micron/hour of exposure-deposition rate

Michael Donley, Program Manager, Wright Patterson- Ceramic bearings
(513) 255-6485

Toshiba- Carbide surface is decarbonized by a hydrogen-oxygen plasmas to remove cobalt binder. Still problem of breakage of diamond film rather than wear failure.
John Herb- Solution is non-trivial

Dr. K. Miyoshi, NASA LeRC, Cleveland, Ohio
(216) 433-6078

Prepare surface 200 angstroms max. roughness. Very difficult to bond to steel. Use silicon nitride substrate or silicon dioxide substrate. Low moisture causes high friction. Would not recommend for MFJ!

Dr. I. L. Singer, Chief of Wear Section, Navy Research Lab.
(205) 767-2327

Apply polycrystalline to both surfaces, if it sticks. Surface is crystalline and will run-in in about 20 cycles. Coefficient of thermal expansion for steel is 6 times higher than it is for diamonds, therefore a compression force is already being applied. Buckling is a function of thickness squared- suggest thickness 3 to 5 microns. Thicker will pop off. Will be just as smooth as original surface. DLC applied to only one surface works quite well if bond can survive thermal expansion.

Hohman Plating, Dayton, Ohio - plating source
(513) 228-2191

Kent Roller. Tribology, Bal Aerospace, Colorado - testing source.
(303) 939-4548

Vitolube, paint and bake, low temperature, Moly Disulfide/ hydrated Moly oxide.
Discovered moisture range of 0 to 100 C exhibited unique friction coefficient.

Dick Gordman

SDI Lockheed selected Mo-disulfide as best friction/wear surface.

Vacuum deposited rotating sphere with rod targets

Doped elemental 3% Nickel, Gold, Tantalum

Gene Lonnette - Sales

Dick is engineer on sputtered film.

Molydisulfide can be sputtered.

Have applied on a few low temperature small precision bearings. No feedback, but also no complaints.

Our 8-inch sphere is quite large, but can do!

Usually 0.5 to 0.75 micron thickness on these small parts. Maximum could go to 2.0 micron if we need it.

We have only 400 or so cycles - don't need extended wear.

Extreme low temperature bonding -420 F.

Will likely have helium gas to purge moisture - could have vacuum.

Telecon 21 April 92

Sputter deposition is recommended process; developed in early 70's. Will use Rod target along centerline. Has vacuum chamber large enough- have not done parts this large (8-inch dia.). Triode is not Dual Ion beam, not familiar with Beam Alloy work. Will dope molydisulfide with about 3% volume for best friction results. Used in satellite work.

7.0 INSULATION

7.1 MECHANICAL FLEX JOINT (MFJ):

The current flex joints on the SSME fuel ducts consists of both inner and outer bellows elements. The space between the two bellows elements contains argon. During ambient temperatures the argon is a gas within the jackets but when it is exposed to liquid hydrogen, it condenses to a liquid and thus creates a partial vacuum between the two jackets. When the bellows warms back to ambient temperatures the argon once again becomes a gas. This "vacuum jacket" insulation prevents ice formation on the outer bellows that would restrict flexibility of bellows convolutions.

Ice formation on the exterior of the MFJ must also be avoided since the seals would be scraped past ice on the matching sealing surface during rotation of the MFJ. Future design activity needs to address this problem and investigate the best method of preventing seal damage due to external ice formation. Several concepts which have been considered during the current study and are the following:

- (a.) External bellows over the MFJ to provide a "vacuum jacket."
- (b.) Flexible "baggy" type jacket with an inert gas purge. This would be similar to the current fix for cryopumping on the Low Pressure Fuel Duct of the SSME.
- (c.) Combination of foam insulation sections joined together by a "baggy" to provide a moisture barrier over the seals area.

No detail design concepts were developed on any of these concepts.

RECOMMENDATION: Suggest the subject of insulating the MFJ be investigated further with the major emphasis on the movable insulation.

7.2 DYNAMIC SEALS TEST FIXTURE (DSTF):

Insulation is a major design issue on the DSTF. The test fixture movable insulation design could provide some very helpful insight on movable insulation for the MFJ. The thermal protection system suggested for the DSTF is to insulate the fixture with a suitable thickness of expanded foam insulation and the movable piston shaft provided with a inert gas barrier to prevent ice formation from causing the piston seals to leak.

The following is a short comment contained in the Thermal section of the DSTF:

How much insulation is going to be needed on the DSTF? This will directly effect the amount of heat gain to the DSTF. Another issue other than the amount and type of insulation is the problem of the movable insulation that will be necessary where the piston shaft enters the plunger. Ice buildup in this area could be catastrophic to the seals reliability on the plunger.

RECOMMENDATION: Suggest the subject of insulating the DSTF be investigated further with the major emphasis on the movable insulation.

8.0 TEST PLANS

There should be a well structured test plan and procedure written for the DSTF and also the MFJ. Currently there hasn't been a test plan written for either of these projects. This is a very critical aspect of the success or failure of this project and will be discussed briefly below.

8.1 TEST PLAN FOR THE DYNAMIC SEALS TEST FIXTURE

The test plan for the DSTF has not been developed at this time. Pending the decision to continue the dynamic seals test program it will be necessary to develop a test plan documenting the test objectives and requirements. There is a copy of the Static Seals Test Plan included in the appendix of this report. The static seals test uses much of the same test equipment as the dynamic seals test and this report is included to possibly help construct a Dynamic Seals Test Plan.

RECOMMENDATION: Write a Dynamic Seals Test Plan when and if the projects progresses to a stage of being built.

8.2 TEST PLAN FOR THE MECHANICAL FLEX JOINT

The test plan for the Mechanical Flex Joint has not been developed at this time. Pending the decision to continue the Dynamic Seals Test program it will be necessary to develop a test plan documenting the test objectives and requirements.

RECOMMENDATION: Write a Mechanical Flex Joint Test Plan when and if the projects progresses to a stage of being built.

9.0 CONCLUSIONS

Several things have been concluded from the design effort to build a MFJ to replace the current flex joint configuration on the Low Pressure Fuel Duct on the SSME.

With the proper design and stress analysis, it has been concluded that the seals would be a single point failure on the MFJ. It became imperative to establish an acceptable leakage rate and then test the seals to make sure they will work within the design environment. The DSTF has been designed to help determine whether seal leakage would be a problem. Several seal vendors have been identified that have full confidence their seals will work for this test program. Although seal leakage would be considered a major problem, it is not believed to be catastrophic on the MFJ.

Originally it was believed that an inline ball joint, to replace the bellows in the fuel duct would be too large. Numerous design concepts were established and parameters defined to establish the envelope of a MFJ. After evaluating Sverdrup's design concepts and parameters it was concluded a MFJ could be designed to conform to the envelope of the current flex joint.

Another problem in the design of the MFJ is the problem of high torque and determination of a method to calculate the torque. Sverdrup established the necessary equations to estimate the necessary torque to flex the joint. Although it was concluded that torque would be driven by the geometrical design of the MFJ, calculation of an acceptable torque could be made.

Sverdrup MFJ design (See Figure 7) has greatly reduced the amount of internal resistance associated with the current bellows "tripod" configuration. The "tripod" design occupies approximately 30% of the internal volume at the bellows but the MFJ has no internal support and therefore has no internal flow resistance. The unrestricted flow in the MFJ should be analyzed and will likely provide a basis for further reduction in size, weight and overall envelope.

The vendor Securamax has insured that they could machine the highly specialized spherical surfaces of the MFJ. Assuming that Securamax can build the MFJ and that the dynamic seals test is successful, the MFJ is believed to be a viable replacement to the current flex joint configuration.

10.0 RECOMENDATIONS

The following section is a summarization of the recommendations made throughout this document. The issues listed below have not been resolved and deserve additional consideration during any additional design process.

2.3.3 Self-Centering of the MFJ: The issue of how to keep a MFJ resting in a self-centering position has not been resolved. This issue must be resolved before a MFJ can replace a flex joints in the current Low Pressure Fuel Duct configuration.

3.0 MFJ Literature/Historical Search: Conduct a literature search to insure some new information on a MFJ has not been recently developed.

4.0 Seals: Investigate other aerospace seal vendors with experience in low temperature dynamic face seals.

5.5 Design Considerations for Dynamic Cryogenic Seals: Study surface finishes and the effect they have on dynamic cryogenic seals.

5.5.1 Stress Analysis (of DSTF): Conduct a formal stress analysis before a CDR package is produced.

5.5.2 Thermal Analysis (of DSTF): Conduct a formal thermal analysis before a CDR package is produced. Also consult MSFC-SPEC-256 and MIL-STD-889 on the relationship between dissimilar metals. The thermal expansion of dissimilar metals is very critical to the success of the DSTF.

5.5.3 Weld Fatigue Analysis (of DSTF): Conduct a weld fatigue analysis before a CDR package is produced.

5.6.1.1 Heaters (on DSTF): In an effort to eliminate another leak path or failure point, weld P/N 96M66993-19 and P/N 96M66993-17 together. This is recommended only if heaters are never used in this test fixture.

5.9 Modified Dynamic Seals Test Fixture: If conducting a variety of seal size tests, suggest considering this design approach.

6.1.3 Process/Equipment, Film Characteristics and MFJ Applications for the three thin film depositions should be investigated more extensively.

6.1.4 Other low friction films should be investigated for MFJ applications. Teflon, ball bearings and polymer films are three candidates that merit further investigation.

7.1 Insulating the MFJ should be investigated further with the major emphasis on the movable insulation.

7.2 Insulating the DSTF should be investigated further with the major emphasis on the movable insulation.

8.1 Test Plan for Dynamic Seals Test Program: Write a Dynamic Seals Test Plan when the project progresses to a stage of building the DSTF.

8.2 Test Plan for Mechanical Flex Joint: Write a Mechanical Flex Joint Test Plan when the project progresses to a stage of building the MFJ.

Investigate the design of a hybrid bellows/flex joint. This design would look like Concept #5 (See Figure 7) with an outer bellows containment jacket. This configuration would remedy the problem of the center tripod cracking, and the outer bellows could be used as an insulation jacket and self-centering mechanism.

ENVELOPE COMPARISON
MFJ CONCEPT 5 / SHORT BELLOWS

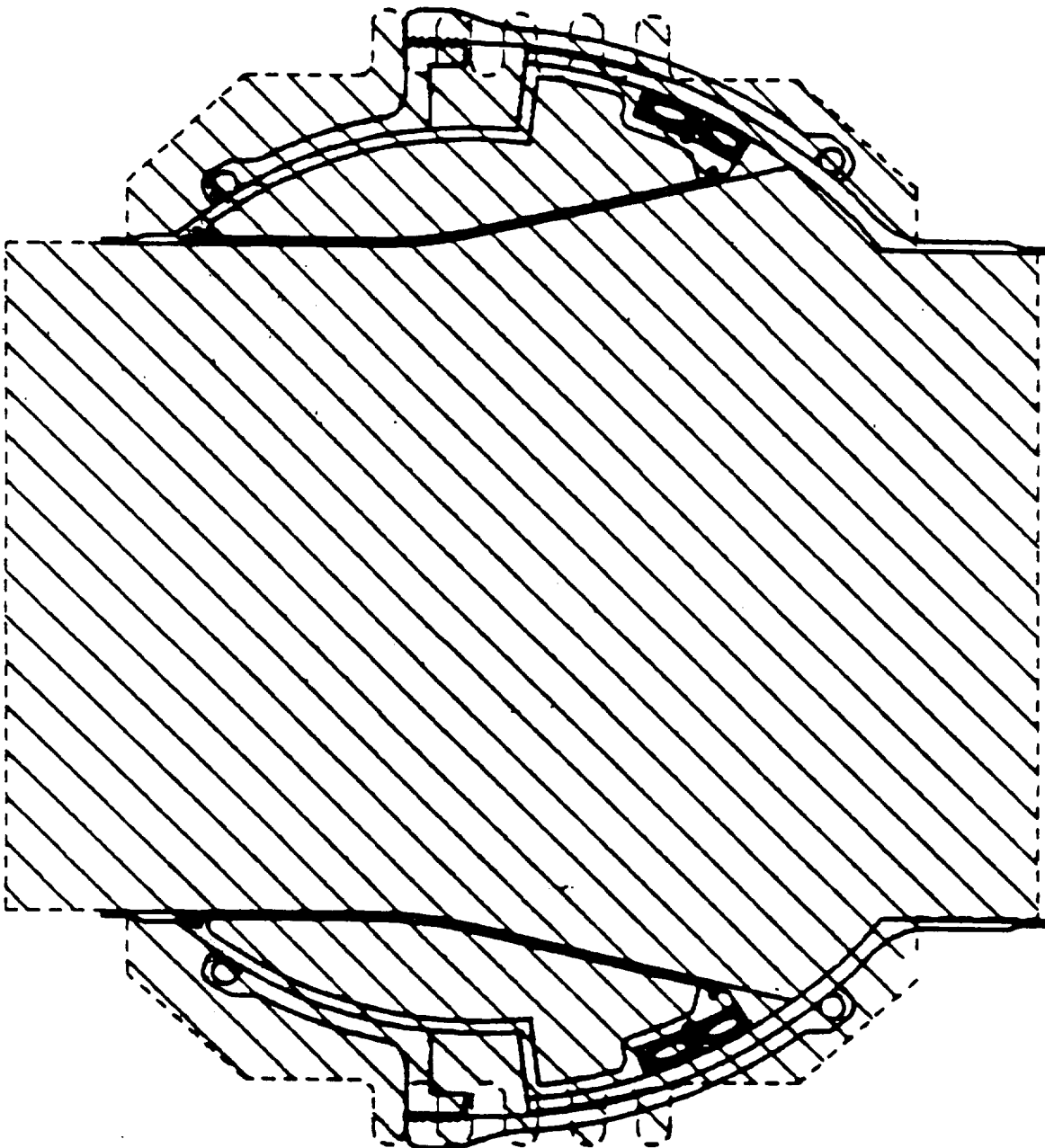


FIGURE 1

SEALING SURFACE V/S BEARING SURFACE

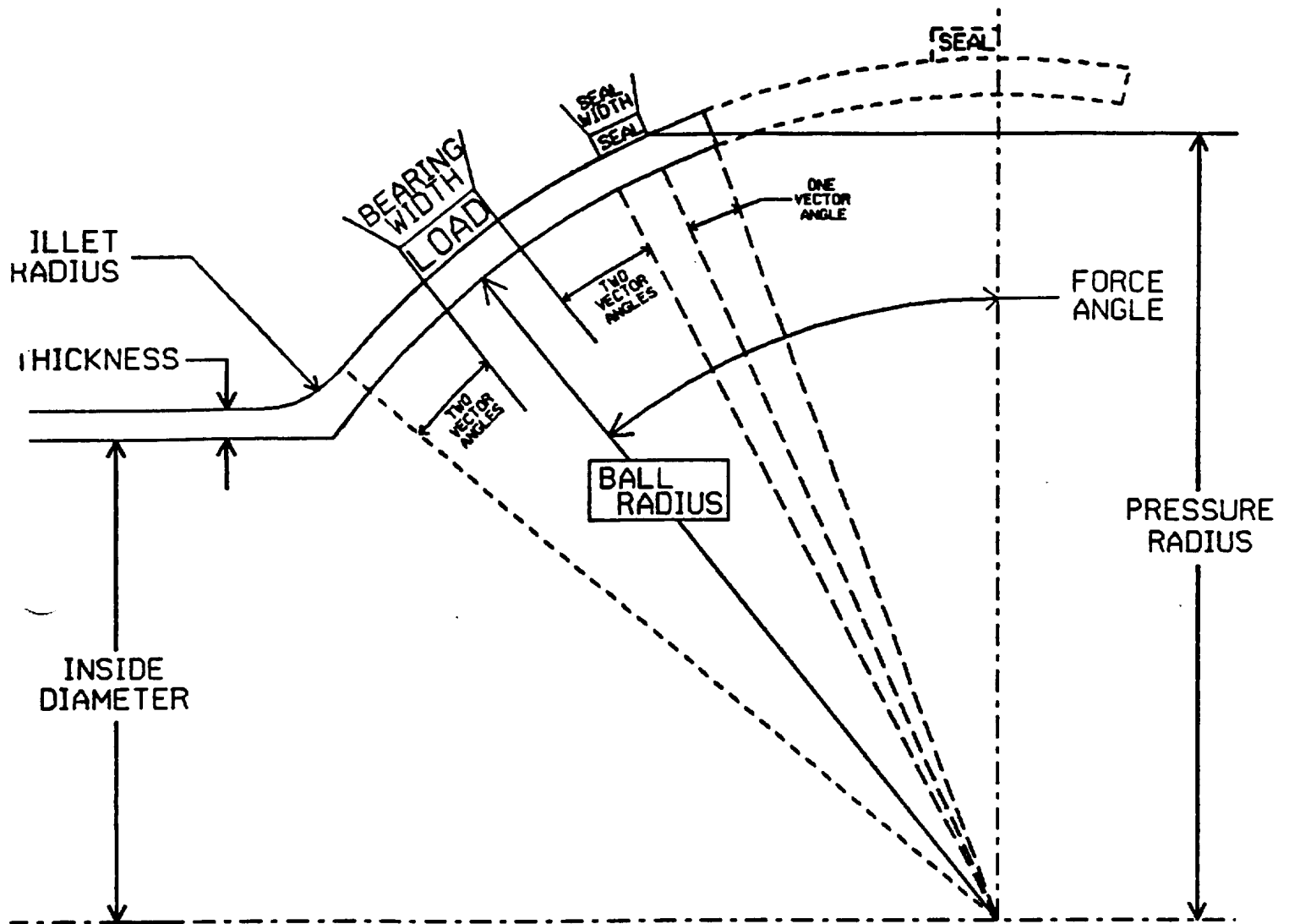
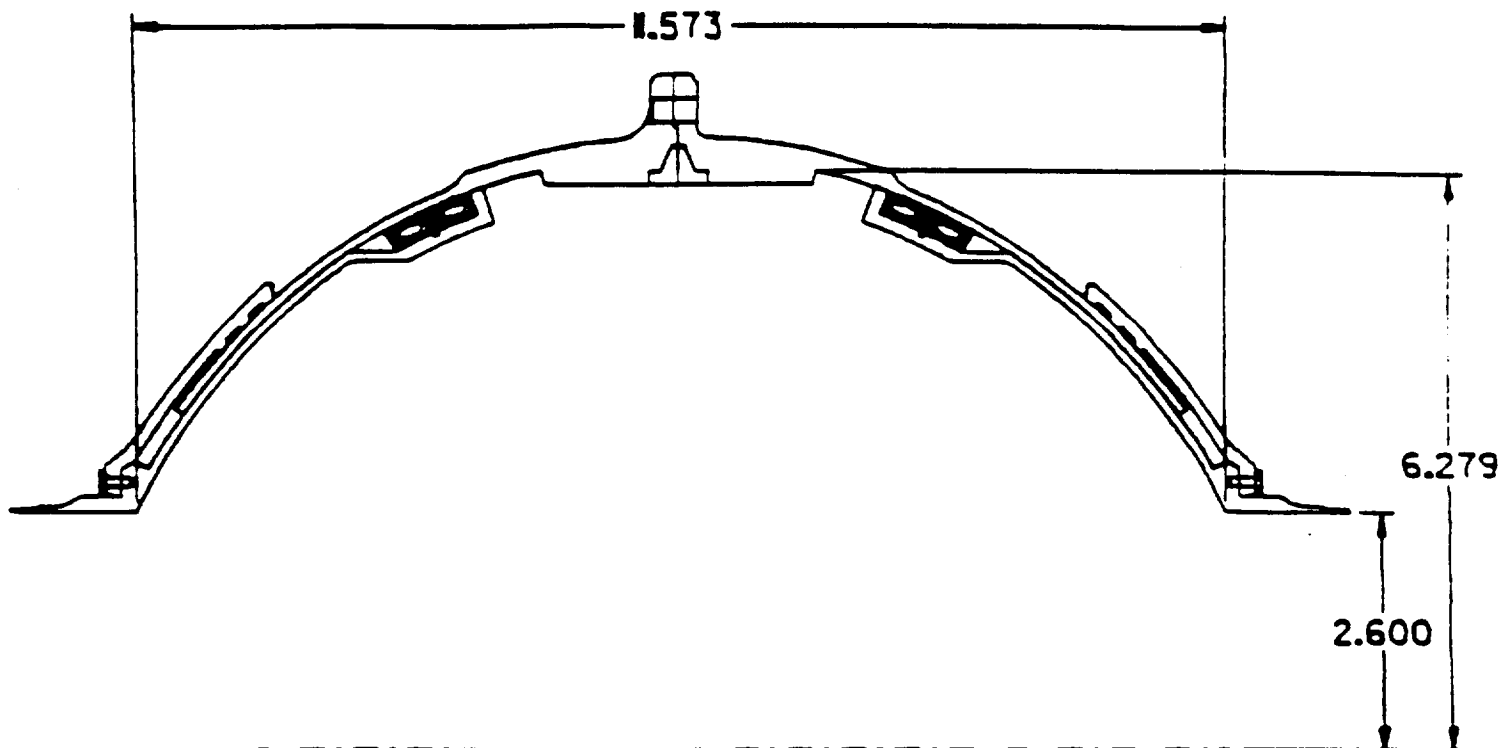


FIGURE 2



CONCEPT 1

TWO SIDES/SINGLE PIVOT

ADVANTAGES:

- 1.) REDUNDANT SEALS
- 2.) ANGULATION - 2 SIDES
- 3.) ROTATIONAL

DISADVANTAGES:

- 1.) TORQUE
- 2.) WEIGHT
- 3.) DIAMETER
- 4.) LENGTH
- 5.) NO FLOW LINER
- 6.) TWO SETS OF SEALS

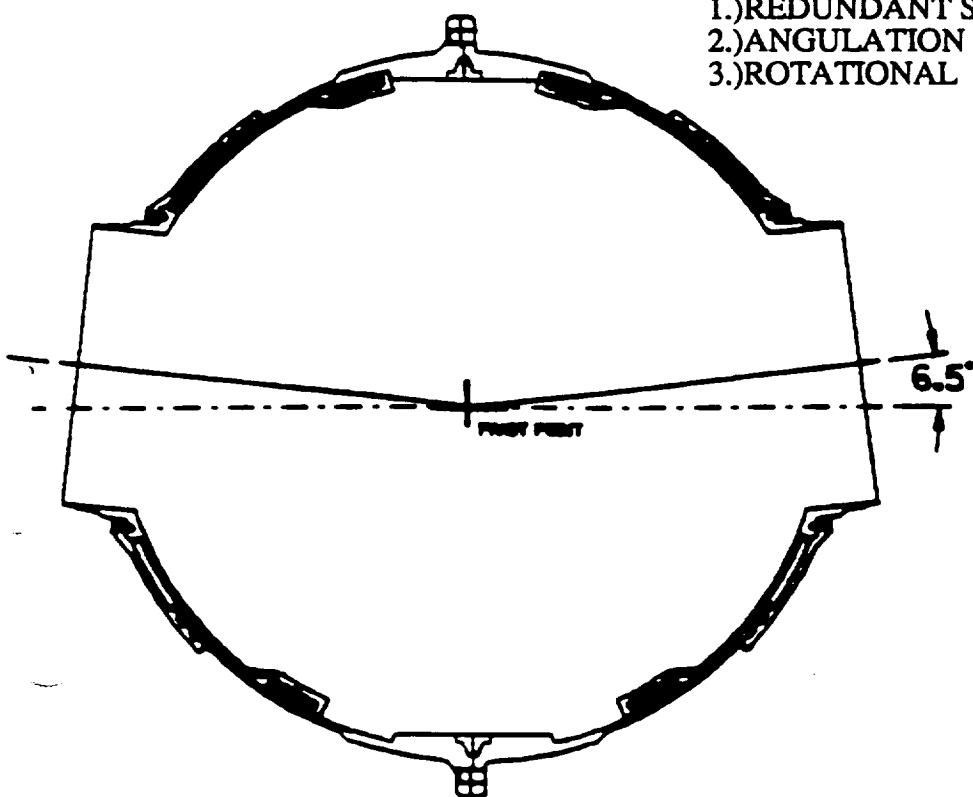
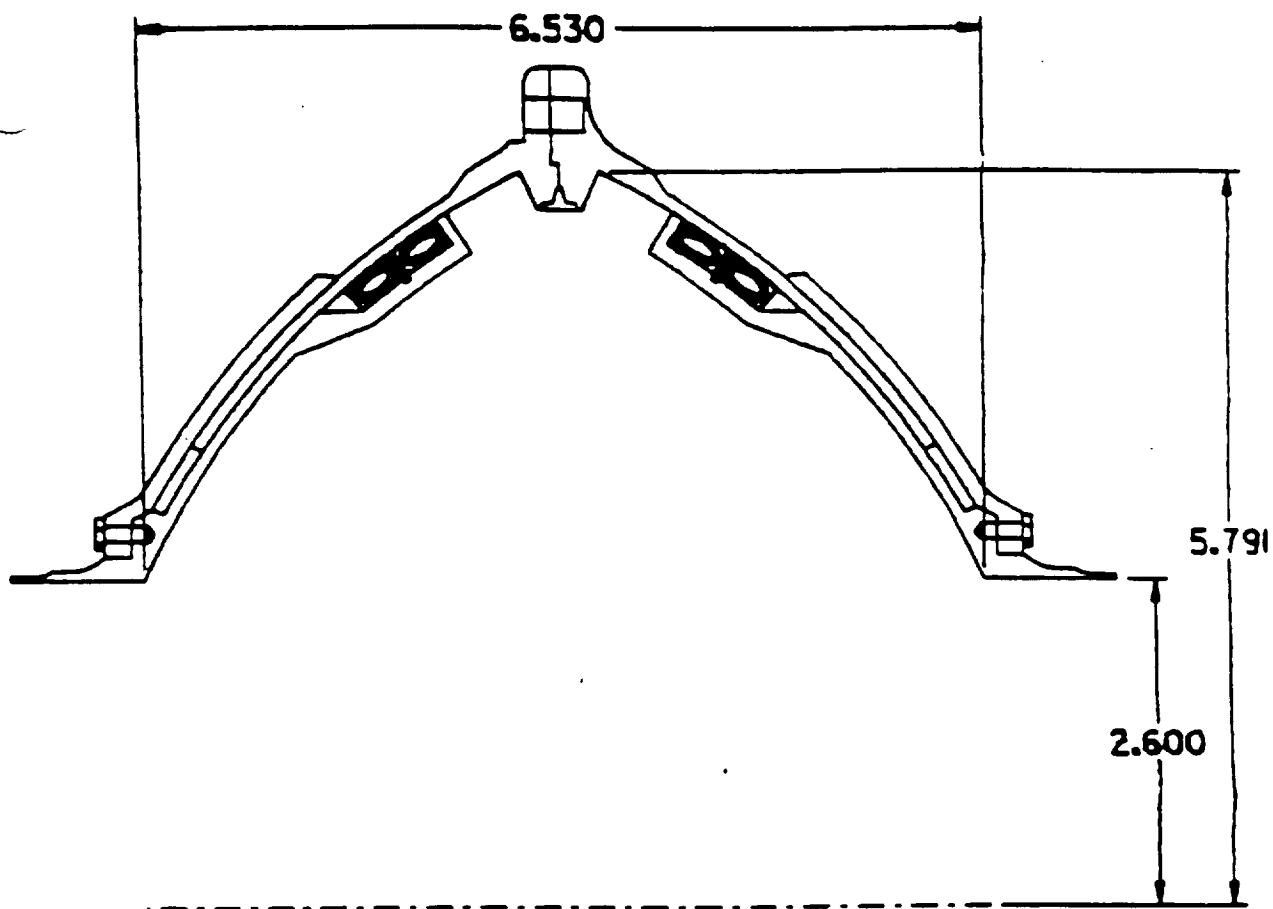


FIGURE 3



CONCEPT 2

TWO SIDES/DUAL PIVOT

ADVANTAGES:

- 1.) REDUNDANT SEALS
- 2.) ANGULATION - 2 SIDES
- 3.) LENGTH
- 4.) WEIGHT
- 5.) ROTATIONAL

DISADVANTAGES:

- 1.) TORQUE
- 2.) NO FLOW LINER
- 3.) DIAMETER
- 4.) SEALS-TWO SETS

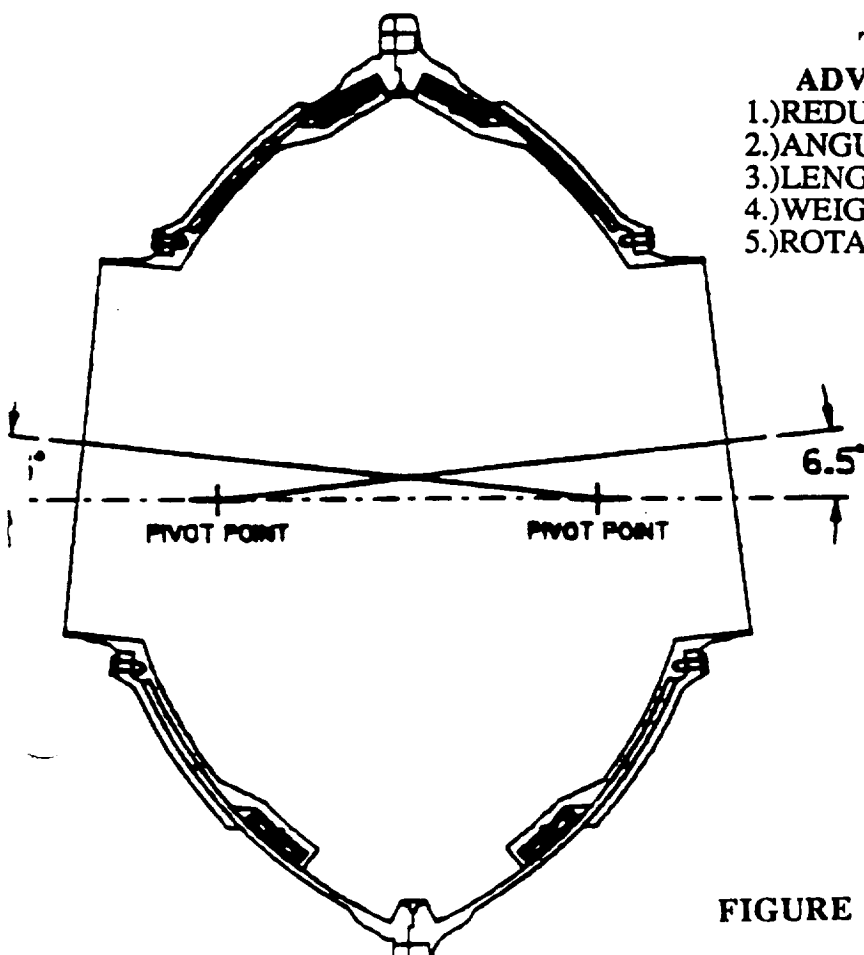
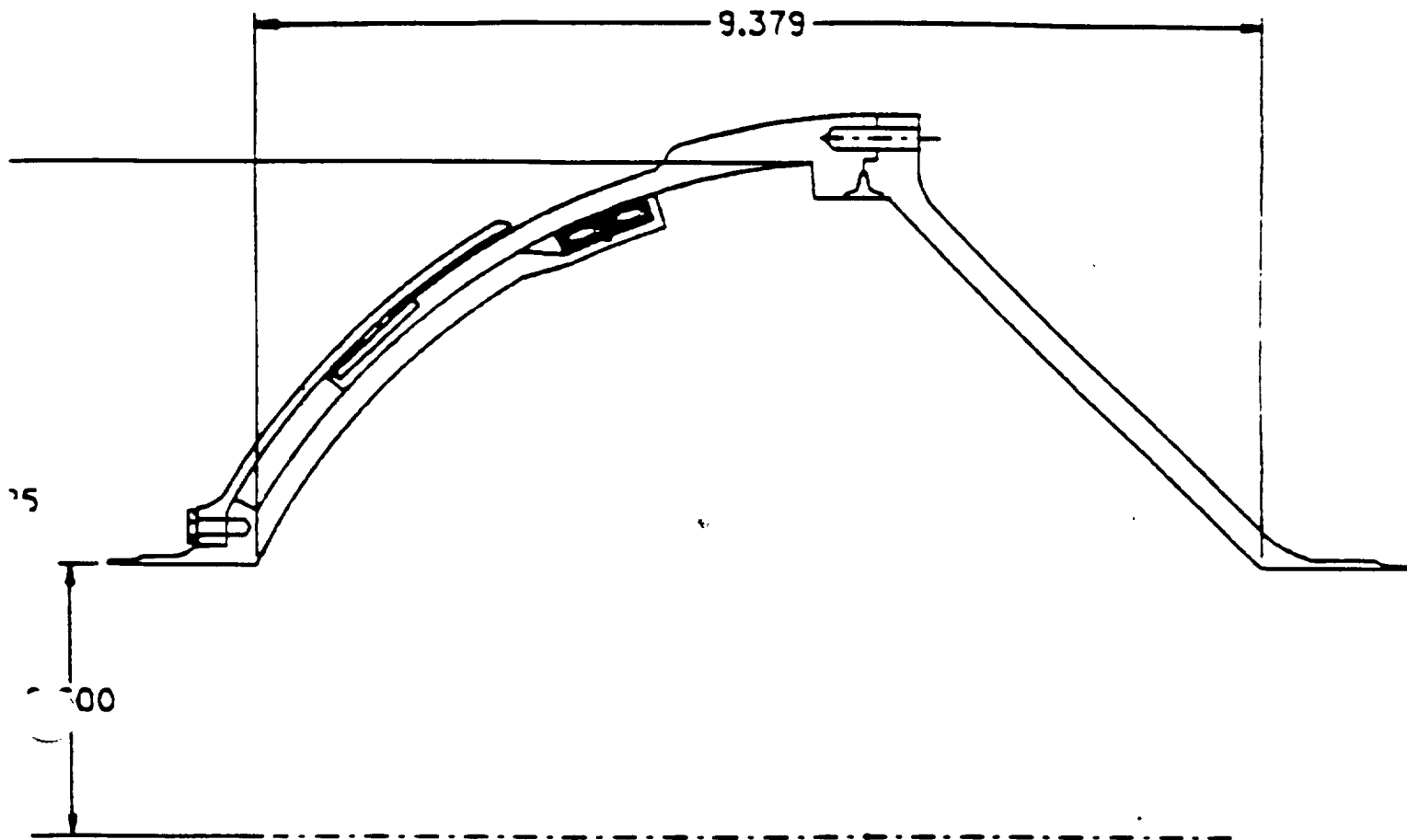


FIGURE 4



CONCEPT 3

ONE SIDES/SINGLE PIVOT

ADVANTAGES:

- 1.) REDUNDANT SEALS
- 2.) SEALS-ONE SET
- 3.) ROTATIONAL

DISADVANTAGES:

- 1.) TORQUE
- 2.) WEIGHT
- 3.) DIAMETER
- 4.) LENGTH
- 5.) NO FLOW LINER
- 6.) ANGULATION-1 END

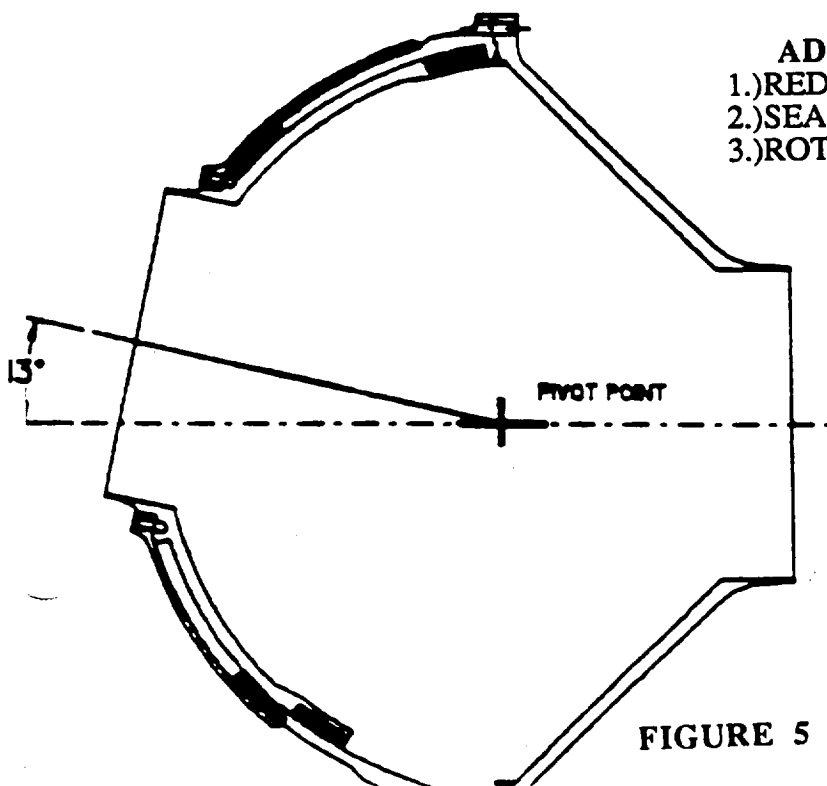
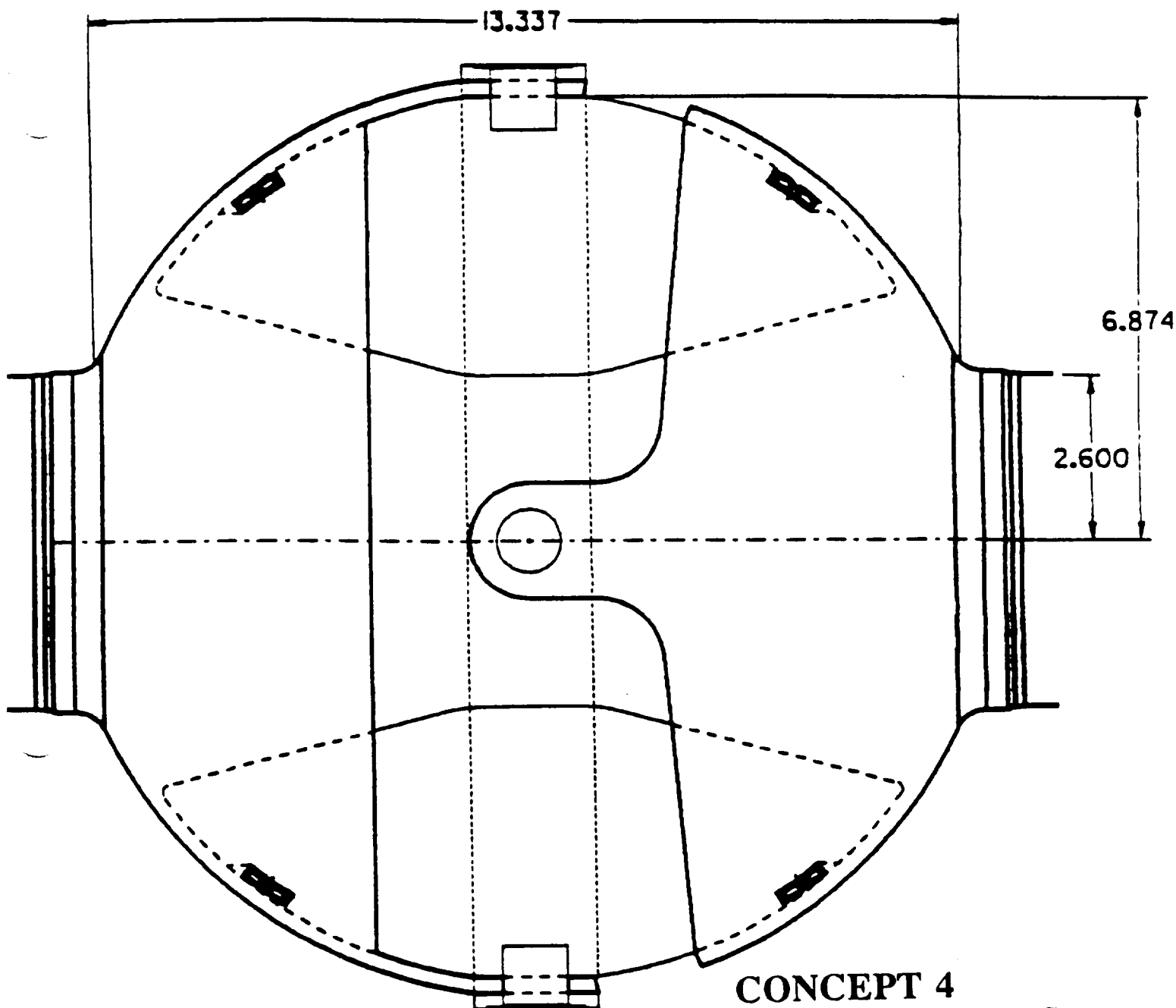


FIGURE 5



CONCEPT 4

TWO SIDES/GIMBAL RING

ADVANTAGES:

- 1.) REDUNDANT SEALS
- 2.) TORQUE
- 3.) FLOW LINER

DISADVANTAGES:

- 1.) NON-ROTATIONAL
- 2.) WEIGHT
- 3.) DIAMETER
- 4.) LENGTH
- 5.) SEALS-TWO SETS

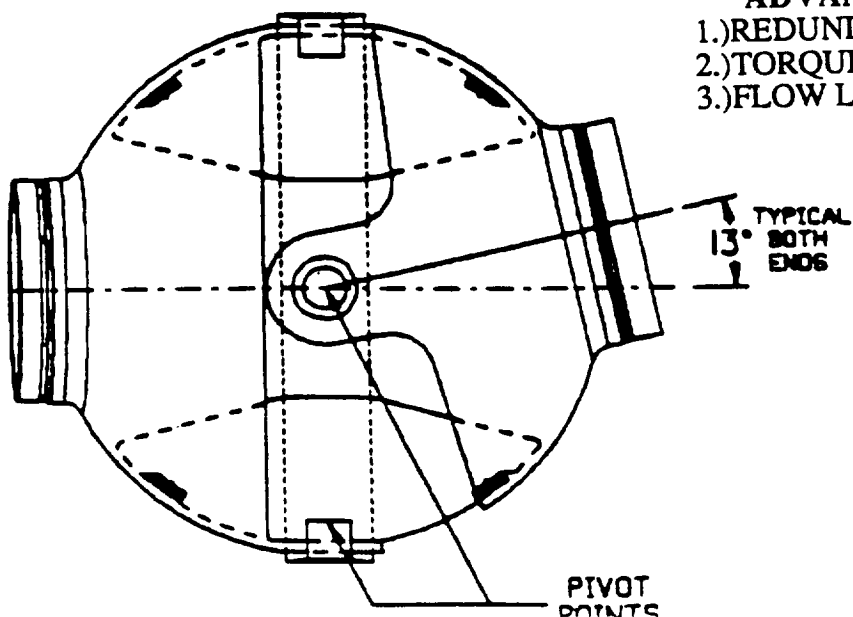
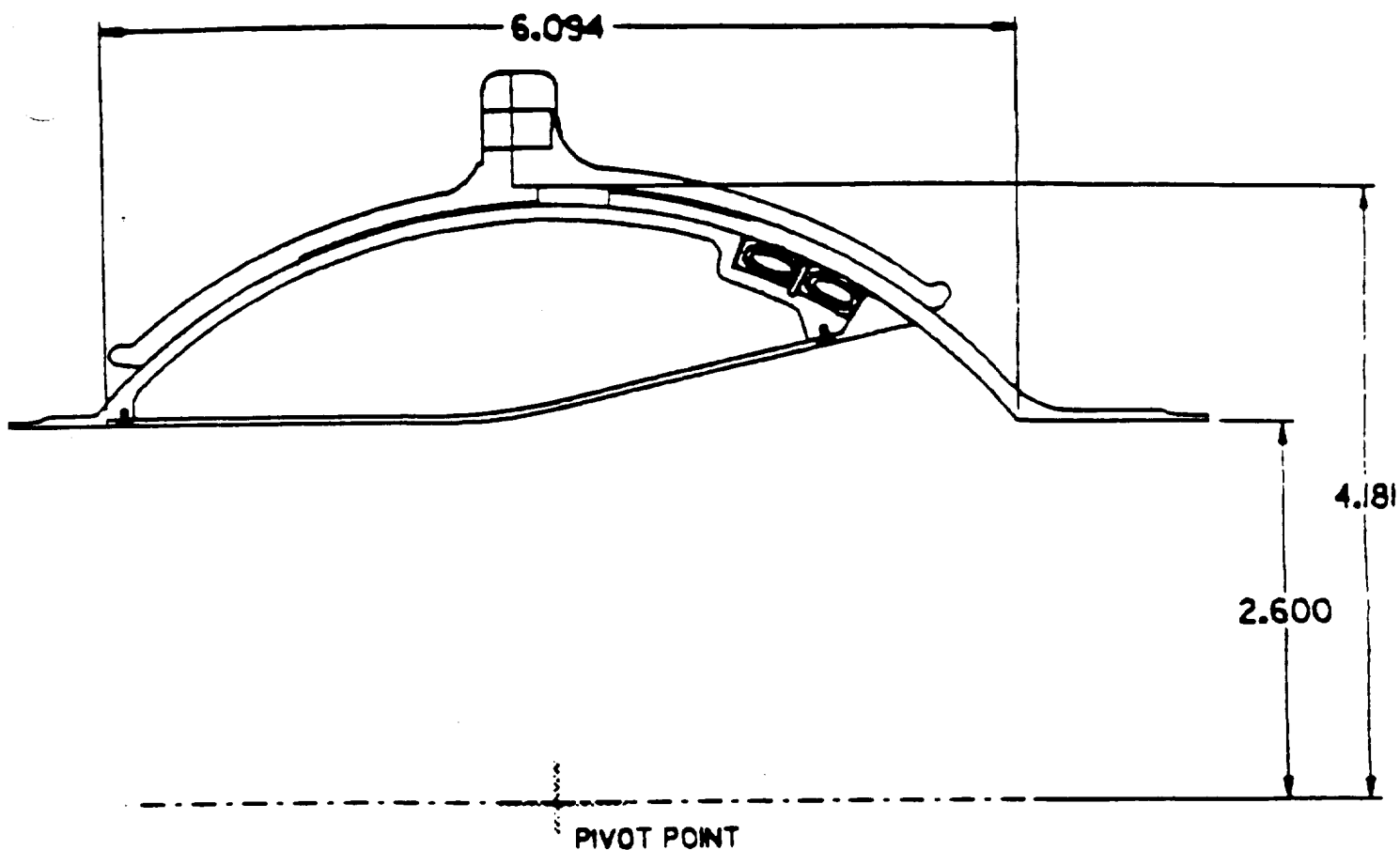


FIGURE 6



CONCEPT 5

TWO SIDES/CONCENTRIC BEARING

ADVANTAGES:

- 1.) REDUNDANT SEALS
- 2.) ANGULATION - 2 SIDES
- 3.) ROTATIONAL
- 4.) LENGTH
- 5.) DIAMETER
- 6.) SEALS-ONE SET
- 7.) FLOW LINER
- 8.) WEIGHT

DISADVANTAGES:

- 1.) TORQUE

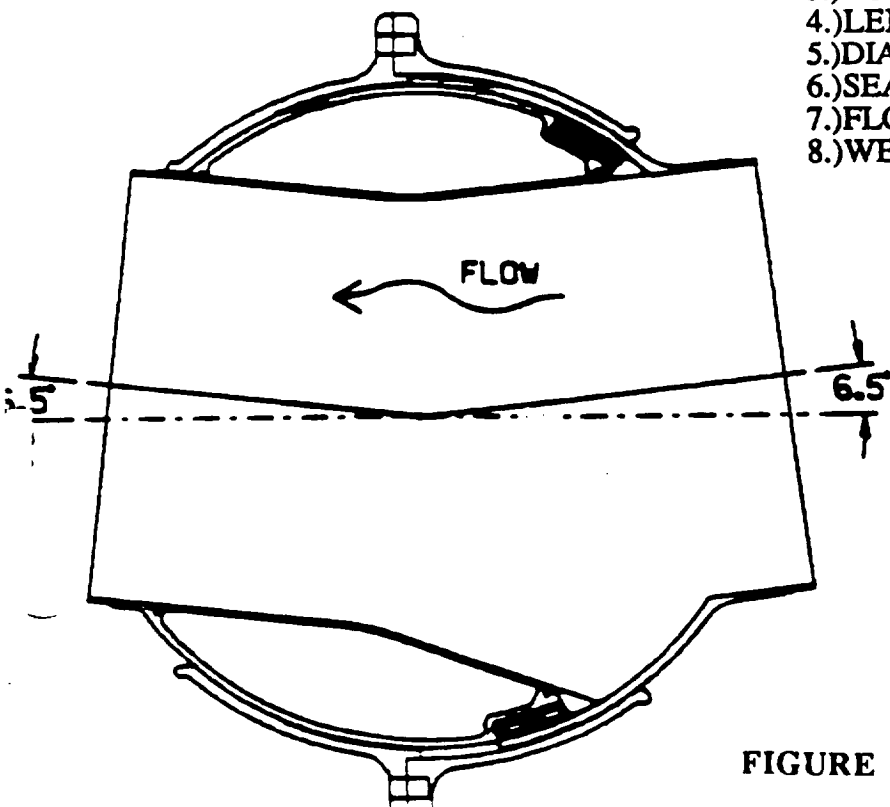
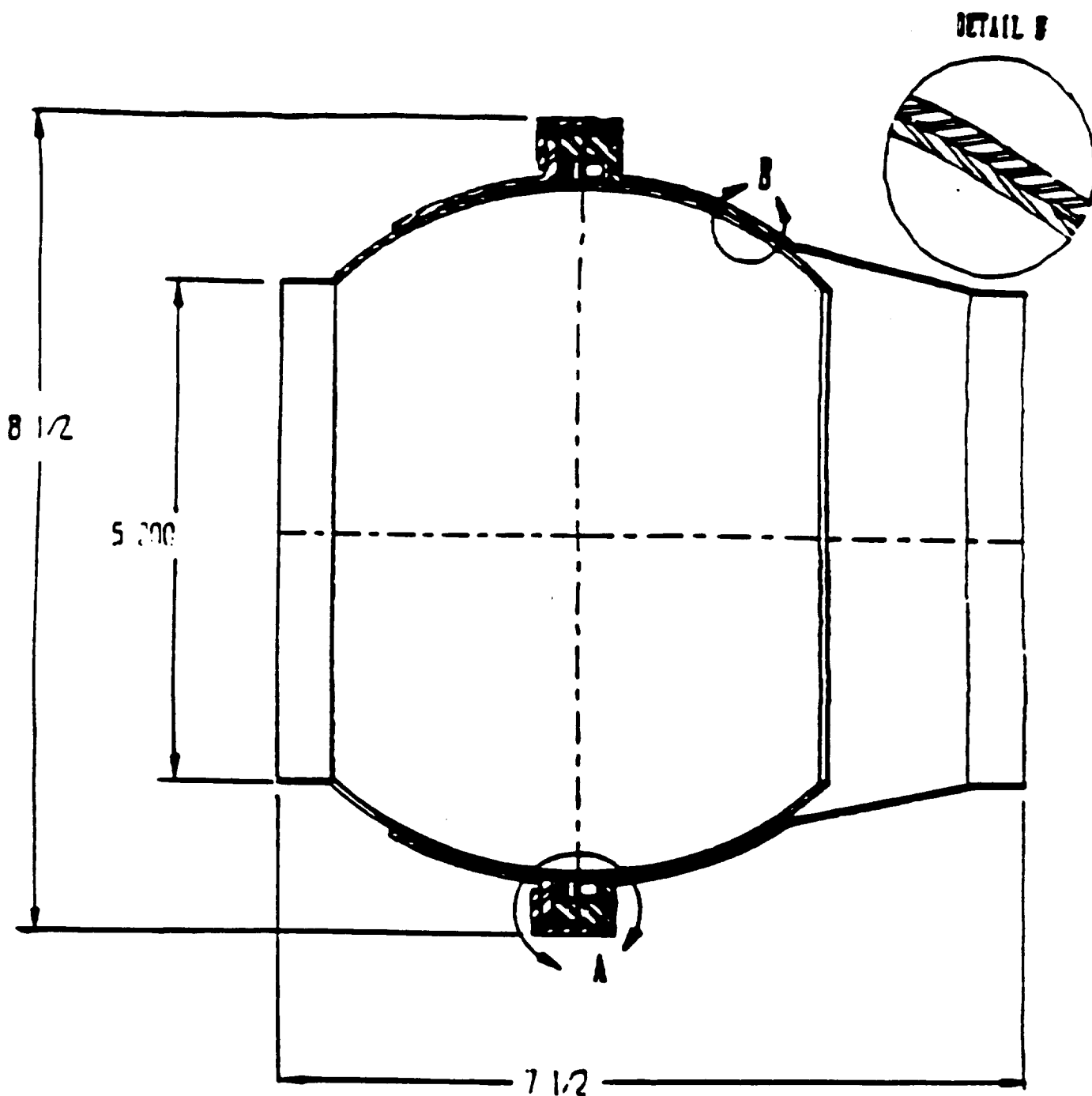
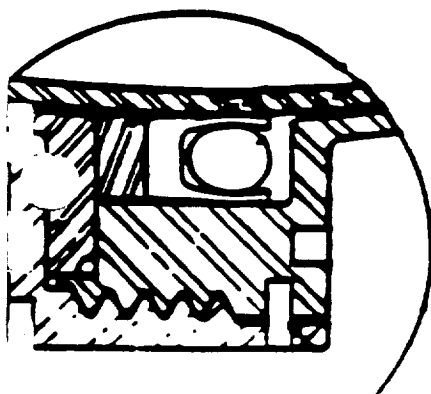


FIGURE 7



DETAIL A



CONCEPT 6 SECURAMAX

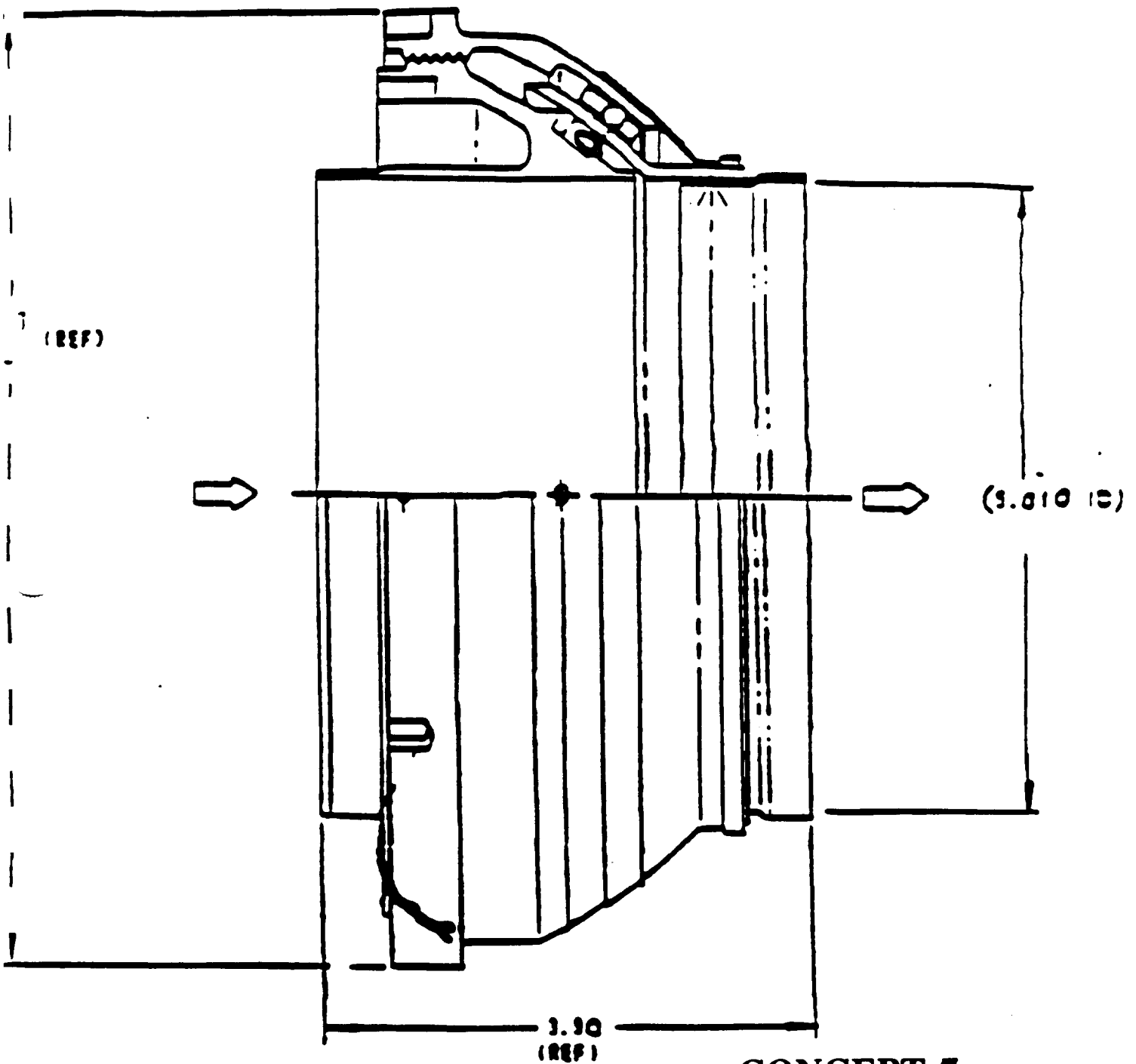
ADVANTAGES:

- 1.) WEIGHT
- 2.) ANGULATION - 2 SIDES
- 3.) ROTATIONAL
- 4.) LENGTH
- 5.) DIAMETER
- 6.) SEALS-ONE SET

DISADVANTAGES:

- 1.) TORQUE
- 2.) NO REDUNDANT SEALS
- 3.) NO FLOW LINER
- 4.) 11 DEGREE ANGULATION

FIGURE 8



CONCEPT 7

STAINLESS STEEL PRODUCTS

ADVANTAGES:

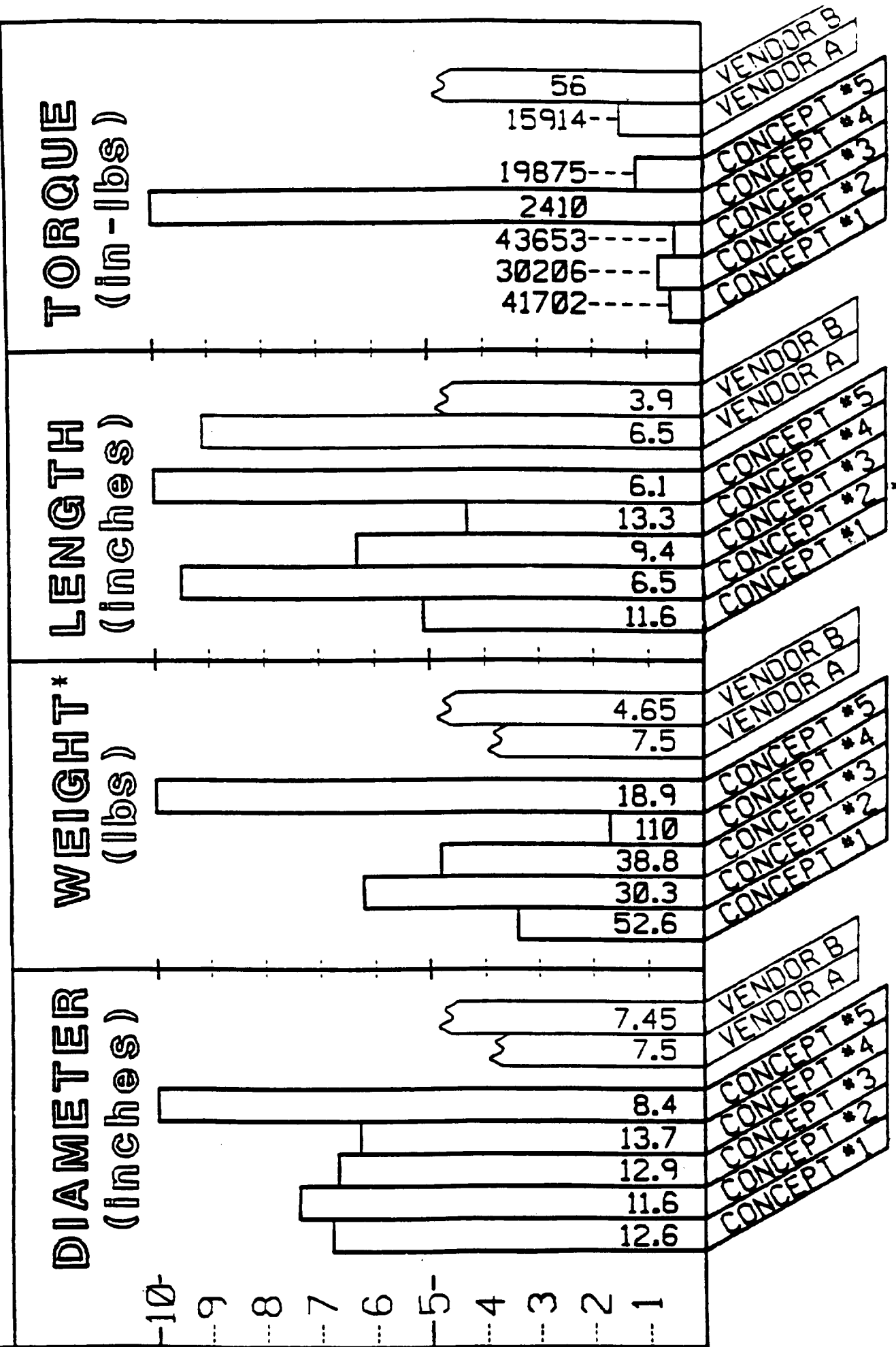
- 1.) TORQUE
- 2.) WEIGHT
- 3.) ROTATIONAL
- 4.) LENGTH
- 5.) DIAMETER
- 6.) SEALS-ONE SET

DISADVANTAGES:

- 1.) NO REDUNDANT SEALS
- 2.) INADEQUATE FLOW LINER
- 3.) 6 DEGREE ANGULATION
- 4.) 250 PSI @ AMB. TEMP.
- 5.) ANGULATION-ONE SIDE

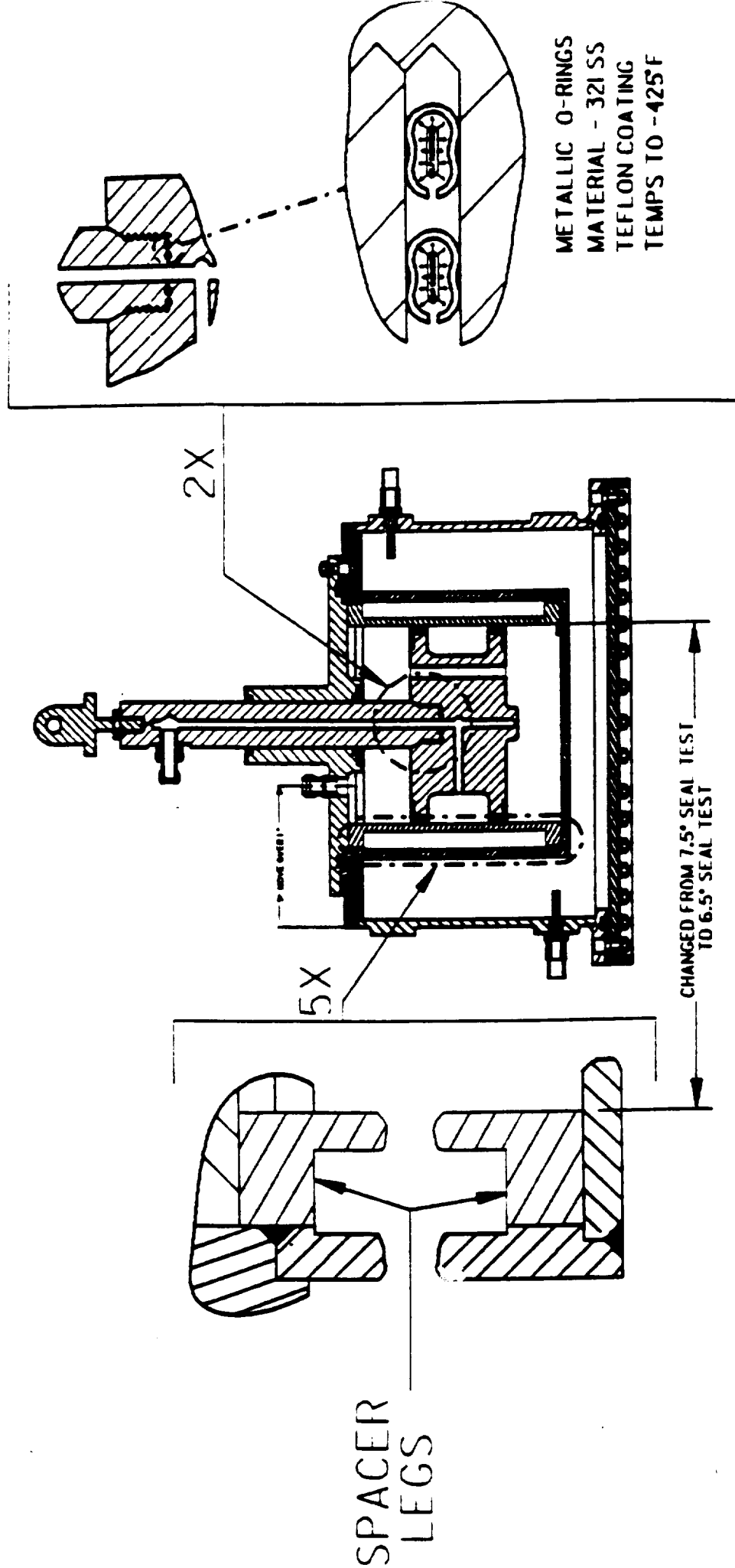
FIGURE 9

MECHANICAL FLEX JOINT RANKING



* EXCLUDING END FLANGES & INSULATION

FIGURE 10



MODIFIED DYNAMIC SEAL TEST FIXTURE

FIGURE 11

APPENDIX A

MECHANICAL FLEX JOINT DESIGN REQUIREMENTS DOCUMENT

DESIGN REQUIREMENTS
FOR
MECHANICAL FLEXIBLE JOINT

SVERDRUP TECHNOLOGY, INC.
MSFC GROUP
PROPULSION DEPARTMENT

SUB-TASK NO. 324-005-01

REVISION 0
JULY 31, 1991

DESIGN REQUIREMENTS FOR
MECHANICAL FLEXIBLE JOINT

CONTENTS

	<u>Page</u>
1.0 SCOPE	1
2.0 APPLICABLE DOCUMENTS	1
3.0 DESIGN REQUIREMENTS	4
3.1 Design Conditions	4
3.2 Performance requirements	6
3.2.1 Pressure Requirements	6
3.2.2 Temperature Requirements	6
3.2.3 Surge Pressure	6
3.2.4 Thermal Cycling	6
3.2.5 Thermal Shock	6
3.3 Physical Requirements	7
3.3.1 Envelope	7
3.3.2 Weight	7
3.3.3 Strength	7
3.3.4 Factors of Safety	7
3.3.5 Surface Finish	7
3.3.5.1 Surface Wear	7
3.3.6 Electrical Continuity	7
3.3.7 Interchangeability	7
3.3.7.1 Design Tolerances	8
3.3.8 Failure Deterrent	8
3.3.9 Minimum Natural Frequency	8
3.3.10 Thermal Insulation	8
3.3.11 Null Position Indication	8
3.4 Environmental Requirements	8
3.4.1 Storage Environment	9
3.4.1.1 Ambient Air Temperature	9
3.4.1.2 Ambient Pressure	9
3.4.1.3 Humidity	9
3.4.2 Ground Handling and Transportation Environment	9
3.4.2.1 Ambient Air Temperature	9
3.4.2.2 Ambient Pressure	9
3.4.2.3 Humidity	9
3.4.2.4 Acceleration	9
3.4.2.5 Vibration	9
3.4.2.6 Mechanical Shock	9

Contents (continued)		<u>Page</u>
3.4.3	Mission Environment	9
3.4.3.1	Temperature	10
3.4.3.2	Pressure	10
3.4.3.3	Acceleration	10
3.4.3.4	Random Vibration	10
3.4.3.5	Mechanical Shock	10
3.5	Construction Requirements	10
3.5.1	Selection of Specifications and Standards	10
3.5.2	Welding	11
3.5.3	Stress Corrosion and Cracking	11
3.5.4	Threaded Fasteners	11
3.5.5	Safety Wiring and Cotter Pinning	11
3.5.6	Drawings	11
3.6	Materials	11
3.6.1	Dissimilar Metals	11
3.6.2	Fungus Nutrient Materials	11
3.6.3	Hazardous Materials	11
3.6.4	Lubricants	11
3.6.5	Fluid Compatibility	12
3.6.6	Protective Finishes	12
3.6.7	Outgassing	12
3.6.8	Castings	12
3.7	Cleanliness	12
3.8	Human Engineering	12
3.9	Safety	12
3.9.1	Redundant Seals	12
4.0	DEVELOPMENT TESTING	12
4.1	Seal Test Unit	12
4.2	MFJ Development Tests	13
4.3	Test Conditions	13
4.3.1	Test Procedure	14
4.3.2	Test Report	15
4.3.3	Test Records	15
4.3.4	Test Tolerances	15
4.3.5	Standard Conditions	15
4.3.6	Accuracy of Measurements	15
4.4	Test Requirements	15
4.4.1	Examination of Product	15

Contents (continued)

	<u>Page</u>
4.4.2 Functional Test	15
4.4.2.1 Pressure Drop Test	15
4.4.2.2 Angular Deflection Torque Test	15
4.4.3 External Leakage Test	15
4.4.4 Random Vibration Testing	15
4.4.5 Proof Pressure Test	16
4.4.6 Burst Pressure Test	16
4.4.7 Weight	16
4.4.8 Cleanliness	16
4.4.9 Environmental Temperature Test	16
4.4.10 Mechanical Shock Test	16
4.4.11 Thermal Cycling Test	16
4.4.12 Thermal Shock Test	17
4.4.13 Electrical Continuity Test	17
4.4.14 Disassembly and Inspection	17

LIST OF TABLES

3-I External Load Requirements	5
--------------------------------	---

LIST OF FIGURES

4-1 Temperature Cycling for Development Test	17
--	----

DESIGN REQUIREMENTS FOR
MECHANICAL FLEXIBLE JOINT

1.0 SCOPE

The purpose of this document is to establish the design requirements for a state-of-the-art Mechanical Flexible Joint. The objective of the Mechanical Flexible Joint (MFJ) is to be utilized as the replacement for the bellows expansion joint in the SSME Low Pressure Fuel Pump Discharge Duct. Design of the MFJ shall not preclude future application in liquid oxygen (LOX) service.

2.0 APPLICABLE DOCUMENTS

The following documents of the issue in effect on the date of this document form a part of these requirements to the extent specified herein. In the event of conflict between documents referenced below and other detail content of this document, the requirements specified herein shall govern. Safety documents are an exception to this rule; they take precedence over other requirements.

SPECIFICATIONS

Military

MIL-B-5087	Bonding, Electrical and Lightning Protection, For Aerospace Systems
MIL-C-45662	Calibration System Requirements
MIL-P-25508	Propellant, Oxygen
MIL-P-27201	Propellant, Hydrogen
MIL-P-27401	Propellant Pressurizing Agent, Nitrogen
MIL-P-27407	Propellant Pressurizing Agent, Helium

MSFC

MSFC-SPEC-164	Cleanliness of Components for Use in Oxygen, Fuel, and Pneumatic Systems
---------------	--

MSFC-SPEC-250	Protective Finishes for Space Vehicle Structures and Associated Flight Equipment, General Specification for
MSFC-SPEC-522	Design Criteria for Controlling Stress Corrosion Cracking
MSFC-SPEC-560	The Fusion Welding of Steels, Corrosion and Heat Resistant Alloys

JSC

SP-R-0022	Vacuum Stability Requirements of Polymeric Material for Spacecraft Application, General Specification
-----------	---

FEDERAL

A-A-50767	Analyzer, Surface Finish
TT-I-735	Isopropyl Alcohol

STANDARDS

Military

MIL-STD-280	Definitions of Item Level, Item Exchangeability, Models and Related Terms
MIL-STD-810	Environmental Test Methods
MIL-STD-889	Dissimilar Metals
MIL-STD-1472	Human Engineering Design Criteria
MIL-STD-1522	Standard General Requirement for Safe Design and Operation of Pressurized Missile and Space Systems
MS33540	Safety Wiring and Cotter Pinning, General Practice for

NASA

MSFC-STD-486	Threaded Fasteners, Torque Limits for
MSFC-STD-506	Materials and Processes Control, Standard

MSFC-STD-509

Lubricant Selection

MSFC-STD-555

MSFC Engineering Documentation
Standard

Industry

ANSI B46.1

Surface Texture (Surface Roughness,
Waviness, and Lay)

OTHER PUBLICATIONS

Handbooks

MIL-HDBK-5

Metallic Materials and Elements for
Aerospace Vehicle Structures

MIL-HDBK-17

Plastics for Aerospace Vehicles

MSFC-HDBK-505

Structural Strength Program
Requirements

MSFC-HDBK-1453

Fracture Control Program
Requirements

NASA Manuals

MM8070.2

Specifications and Standards,
Approved Baseline List

NHB-8060.1

Flammability, Odor, and Offgassing
Requirements and Test Procedures for
Materials in Environments That
Support Combustion

NSTS 07700,
Volume X

Space Shuttle Flight and Ground
System Specification

Technical Memorandum

NASA TM-86538

Design and Verification Guidelines
for Vibroacoustic and Transient
Environments

Rocketdyne

RSS-8561-24

SSME Structural Loads Criteria

Rockwell

ICD-13M15000

Space Shuttle Orbiter Vehicle/Main
Engine Interface Control Document

3.0 DESIGN REQUIREMENTS

3.1 Design Conditions. The design conditions for the MFJ are as follows:

Service: SSME Low Pressure Fuel Discharge Duct
expansion joint bellows replacement

Maximum Design Pressure (MDP): 343 psia

Maximum Operating Pressure (MOP): 326 psia

Design Temperature: -415 °F

Design Flowrate: 163 lbm/sec of LH₂

Pressure Drop: 10 psid maximum at design LH₂ flowrate
and design angular deflection

Deflection Torque: 7355 in-lb maximum at 11.5 degrees
angular deflection with pressure of
326 psia and temperature of -415 °F.

5019 in-lb maximum at 11.5 degrees
angular deflection for ambient
pressure and ambient temperature.

(Note - These torques are design
goals based on the current long flex
joint requirements.)

Duct Size: 5.20" I.D. X 5.264" O.D. (0.032" wall)

End Connection Type: Flanges

Length: TBD Maximum

Design Angular Deflection: ± 13.0 degrees omni-
directional

Cycle Life: (a) Full angular deflection cycles of
(i) 400 at design pressure and
temperature conditions; and
(ii) 2800 non-operational (at ambient
temperature and pressure)
(b) Thermal shock from ambient
temperature to -415 °F with LH₂
flowrate of TBD lb/sec and return to
ambient temperature for 20 cycles.

External Loads: Per Table 3-I.
(Note - External loads are based on
RSS-8561-24 for the LPFTP Discharge
Duct worse case flex joint location.)

Table 3-I External Load Requirements

Source of Load (Note 1)	Freq. (Hz) (Note 2)	Vibra. Input Axis (Note 3)	Design Loads (Note 4)			
			Axial (lb)	Shear (lb)	Torque (in-lb)	Bending (in-lb)
Displacement (Note 5)			370	540	9,900	19,700
Acceleration			420	340	6,500	3,200
Flow Momentum			1,310			
Vibration 3 σ Random	92	X	350	260	1,400	1,300
	87	Y	520	210	570	1,100
	53	Z	900	220	500	1,100
Superimposed Sine		X	40	90	20	150
		Y	50	40	60	340
		Z	40	30	70	300
Decaying Sine (Transient)		X	230	510	3,700	7,700
		Y	190	70	260	1,800
		Z	400	720	3,700	10,700

- Notes: 1. These loads exclude pressure separating loads.
2. Vibration frequency is based on current SSME LPFTP Discharge Duct configuration with flexible joints.
3. Vibration input axis is referenced to SSME coordinate system.
4. Design loads are referenced to local coordinate system on MFJ.
5. Displacement loads include installation misalignment, engine gimbaling, and thermal expansion/contraction.

External Leakage: Shall not exceed 1×10^{-3} scc/sec GHe when pressurized to the maximum design pressure with Helium for temperatures ranging from -415 °F to +140 °F

3.2 PERFORMANCE REQUIREMENTS

3.2.1 Pressure Requirements. The MFJ shall meet the following pressure requirements:

- (a) The Maximum Design Pressure (MDP). The maximum design pressure of the MFJ shall be 343 psia (358 psig) on the inlet and outlet connections.
- (b) Proof Pressure. The MFJ shall be subjected to a proof pressure of 395 ± 8 psig with LN₂ at a temperature of -305 ± 15 °F at the design angular deflection of 13.0 degrees. Maintain the proof pressure for a minimum of 1 minute and reduce the pressure to 0 psig. Repeat the proof pressure cycle four (4) more times.
- (c) Calculated Burst Pressure. The MFJ shall be designed to withstand a calculated burst pressure of 610 psig at -305 °F without structural collapse or rupture for 2 minutes. The MFJ shall not be required to operate after the application of the calculated burst pressure, but shall meet the external leakage requirements from the MFJ internal cavity to the exterior.

3.2.2 Temperature Requirements. The MFJ shall meet all the performance requirements herein over a design temperature range of -415 °F to +140 °F.

3.2.3 Surge Pressure. The MFJ shall meet the requirements herein after exposure to a surge pressure of 50 psi for TBD ms maximum at either the upstream and/or downstream ports.

3.2.4 Thermal Cycling. The MFJ shall show no performance degradation during and after being subjected to a minimum of five (5) thermal cycles between the acceptance test limits of -415 °F to +140 °F.

3.2.5 Thermal Shock. The thermal shock requirements are TBD. The MFJ design shall be suitable for temperature gradients which occur during cryogenic filling operations in both the vertical and horizontal orientations of the MFJ assembly center-line.

3.3 PHYSICAL REQUIREMENTS

3.3.1 Envelope. The external dimensions of the MFJ shall be such that, in the installed condition, the SSME envelope dimensions per ICD-13M15000 are not exceeded (design goal only).

3.3.2 Weight. The MFJ assembly dry weight shall not exceed 50 pounds.

3.3.3 Strength. The MFJ shall have sufficient strength and stiffness at the design temperature to withstand limit loads and pressure without loss of operational capability for the life of the MFJ and to withstand proof loads and pressures at design temperatures without functional failure. The MFJ shall also comply with the strength requirements of MSFC-HDBK-505 and MSFC-HDBK-1453. Metallic material properties shall be in accordance with MIL-HDBK-5 using "A" basis values for strength calculations. Plastic material properties shall be in accordance with MIL-HDBK-17.

3.3.4 Factors of Safety. The following factors of safety are minimum and shall be used in addition to vibration amplification factors and other safety factors relating to stress.

- (a) Yield Factor of Safety - The yield factor of safety shall be 1.26 on the maximum design pressure.
- (b) Ultimate Factor of Safety - The ultimate factor of safety shall be 1.81 on the maximum design pressure on the body.

(Note - Yield and ultimate factors of safety given above are based on the current LPFTP Discharge Duct design.)

3.3.5 Surface Finish. The surface finish used on sealing surfaces shall be designated in accordance with ANSI B46.1

3.3.5.1 Surface Wear. The wear and attendant particle generation at any dynamic interfacing surfaces (contacting surfaces under relative motion) of the MFJ shall not introduce contaminant into the fluid flow path and shall not impair the function of the specific interface, the MFJ as a whole, or the SSME.

3.3.6 Electrical Continuity. The MFJ shall be electrically continuous over its entire surface and shall comply with the bonding requirements of MIL-B-5087, Class S.

3.3.7 Interchangeability. Mechanical interchangeability in accordance with MIL-STD-280 shall exist between like assemblies, subassemblies and replaceable parts. The

substitution of such like assemblies, subassemblies and replaceable parts shall be easily affected without physical modification to any part of the MFJ, including mounting.

3.3.7.1 Design Tolerances. Provisions shall be made for design tolerances such that items having the dimensions and characteristics permitted by the item drawings are interchangeable without selection or departure from the specified equipment performance.

3.3.8 Failure Deterrent. The MFJ design shall incorporate the following:

- (a) Avoidance of blind pockets where hidden corrosion could develop.
- (b) Provisions for failure propagation protection such that transient out-of-tolerance conditions or component failures will not cause other component or subsystem failures.
- (c) Positive locking on threaded parts and fasteners to prevent loosening during service.
- (d) Use of nonsymmetry of configuration, different connecting sizes, or a comparable means of preventing backward or other improper installation of unidirectional components or piece parts.
- (e) Minimization of sliding fits to prevent frictional failures and/or binding.

3.3.9 Minimum Natural Frequency. The MFJ shall be designed such that the minimum natural frequency is greater than 650 Hz in any direction.

3.3.10 Thermal Insulation. The MFJ design shall be suitable for the addition of TBD inch thickness of thermal insulation (provided by others) or shall be covered by a flexible vacuum jacket.

3.3.11 Null Position Indication. The MFJ shall have suitable markings on the body to provide indication of the null position.

3.4 Environmental Requirements. The MFJ shall be designed to perform during and/or after exposure to any single or reasonable combination of natural and induced/operational environments for pre-launch checkout and test, launch, boost, ascent, STS orbital mission, landing, storage and handling operations.

3.4.1 Storage Environment. The MFJ in a packaged state shall meet the requirements of this document after exposure to any combination of the following storage environments defined herein for a 6 year storage period. The packaged ambient external environment is as specified below.

3.4.1.1 Ambient Air Temperature. The ambient air temperature shall be from -20 °F to +140 °F for sustained periods.

3.4.1.2 Ambient Pressure. The ambient pressure will vary between 31.3 inches Hg (sea level) and 28.0 inches Hg (6,000 feet).

3.4.1.3 Humidity. The relative humidity shall be uncontrolled and range from 0 to 100 percent.

3.4.2 Ground Handling and Transportation Environment. The MFJ mounted on the SSME and in the approved unit packaging shall meet the requirements of this document after exposure to any combination of the following ground handling and transportation environments.

3.4.2.1 Ambient Air Temperature. The ambient temperature of the air external to the shipping container will range from -40 °F to +140 °F.

3.4.2.2 Ambient Pressure. The pressure will vary between 31.3 inches Hg (sea level) and 3.5 inches Hg (50,000 feet).

3.4.2.3 Humidity. The relative humidity will be uncontrolled and range from 0 to 100 percent with condensation in the form of water or ice external to the shipping container. Humidity within the shipping container shall be controlled such that no condensation or frost occurs on the MFJ assembly.

3.4.2.4 Acceleration. The maximum steady-state acceleration shall be 3.5 g's (limit) in any direction.

3.4.2.5 Vibration. When packaged or otherwise prepared for shipment, the MFJ shall withstand the vibration environments specified in MIL-STD-810, Method 514.3, Procedure I.

3.4.2.6 Mechanical Shock. The shock levels into the MFJ shall be controlled by design of the handling and shipping container. The packaged MFJ shall withstand the shock environment of MIL-STD-810, Method 516.3, Procedure II.

3.4.3 Mission Environment. The MFJ shall meet the requirements of this document during and after exposure to the following natural and induced environments over 50 mission

cycles. A mission cycle is defined as the pre-launch checkout and test, launch, orbital operations, and return to ground of the STS.

3.4.3.1 Temperature. The mission temperature environment shall be within TBD °F to TBD °F.

3.4.3.2 Pressure. The mission environmental pressure shall be within 31.3 inches Hg (sea level) to TBD torr.

3.4.3.3 Acceleration. The MFJ shall meet the requirements of this document after exposure to the following acceleration levels for TBD minute duration.

(a) ± TBD g's parallel to the flow axis

(b) ± TBD g's perpendicular to the flow axis

3.4.3.4 Random Vibration. The MFJ shall meet the requirements of this document after exposure to the following random vibration environments applied in each of the three mutually perpendicular axis.

<u>Frequency</u>	<u>Random Vibration</u>	<u>Criteria</u>
20 Hz	@ 0.00300	g^2/Hz
20-50 Hz	@ + 8.3	dB/oct
50-500 Hz	@ 0.03700	g^2/Hz
500-2000 Hz	@ -9.8	dB/oct
2000 Hz	@ 0.00040	g^2/Hz

Composite = 4.99 g_{rms}

Duration: TBD seconds per axis

3.4.3.5 Mechanical Shock. The shock level environments applied along each MFJ coordinate axis are TBD.

3.5 CONSTRUCTION REQUIREMENTS

3.5.1 Selection of Specifications and Standards. All materials, parts, and processes shall be defined by standards and specifications, selected from those of government and industry in accordance with MM8070.2.

3.5.2 Welding All welding shall be in accordance with MSFC-SPEC-560 and weld inspection methods shall meet the requirements of MSFC-STD-506.

3.5.3 Stress Corrosion and Cracking. All metallic materials used in construction of this MFJ shall meet the requirements of MSFC-STD-522.

3.5.4 Threaded Fasteners. Threaded fasteners shall comply with MSFC-STD-486.

3.5.5 Safety Wiring and Cotter Pinning. Safety wiring and cotter pinning shall be in accordance with MS33540.

3.5.6 Drawings. Drawings shall be in accordance with MSFC-STD-555.

3.6 Materials. All materials and material processes shall be in accordance with MSFC-STD-506 and MSFC-SPEC-522.

3.6.1 Dissimilar Metals. Protection of dissimilar metal combination shall be in accordance with MSFC-SPEC-250 and MIL-STD-889. The worst case environment anticipated for the MFJ assembly, shall be considered.

3.6.2 Fungus Nutrient Materials. Materials that are nutrient for fungus shall not be used.

3.6.3 Hazardous Materials. Flammability, toxicity, and fluid compatibility provisions of NHB-8060.1 for both liquid hydrogen (LH₂) and liquid oxygen (LOX) services shall be used in the design.

3.6.4 Lubricants. Lubricants shall be in accordance with MSFC-STD-509.

3.6.5 Fluid Compatibility. Propellant contact surfaces inside the MFJ shall be compatible with the operating fluids and the fluids used to clean the MFJ. The fluids shall be limited to the following:

- (a) Liquid Hydrogen (LH₂) per MIL-P-27201
- (b) Liquid Oxygen (LOX) per MIL-P-25508
- (c) Gaseous Helium (GHe) per MIL-P-27407, Type 1
- (d) Gaseous Nitrogen (GN₂) per MIL-P-27401
- (e) Distilled and deionized water
- (f) Isopropyl alcohol (IPA) per TT-I-735, Grade A.
- (g) Cleaning fluids specified in MSFC-SPEC-164.

3.6.6 Protective Finishes. The provision of MSFC-SPEC-250 shall apply to materials susceptible to corrosion from exposure to the environments.

3.6.7 Outgassing. Materials shall be suitable for use in a space environment, exhibit low outgassing characteristics per SP-R-0022 and shall maintain their mechanical, physical, and electrical properties.

3.6.8 Castings. Castings shall not be used for any pressure retaining parts of the MFJ without MSFC approval.

3.7 Cleanliness. All pressurized elements shall be thoroughly cleaned. Particle contamination test and non-volatile residue (NVR) test shall be in accordance with MSFC-SPEC-164 for oxygen service.

3.8 Human Engineering. The MFJ shall be designed to achieve reliability, maintainability, and safety within the human engineering interface between personnel and equipment. Where applicable, the design shall satisfy the criteria of MIL-STD-1472.

3.9 Safety. The MFJ design shall be in accordance with and meet the requirements of NSTS 07700, Volume X.

3.9.1 Redundant Seals. The MFJ shall have redundant seals of all external leakage paths. The primary and secondary seals shall be independently verifiable by the use of leak check ports or similar methods.

4.0 DEVELOPMENT TESTING

4.1 Seal Test Unit. A full scale development test unit of the sealing features for the MFJ shall be used initially to demonstrate satisfactory seal performance.

The following tests, as a minimum, shall be performed on the seal test unit in the sequence shown below:

- | | |
|--------------------------|---------------|
| 1. External Leakage Test | Para. 4.4.3 |
| 2. Torque Test | Para. 4.4.2.2 |
| 3. Thermal Shock Test | Para. 4.4.12 |
| 4. Random Vibration Test | Para. 4.4.4 |
| 5. Mechanical Shock Test | Para. 4.4.10 |
| 6. External Leakage Test | Para. 4.4.3 |

- | | |
|-------------------------------|---------------|
| 7. Torque Test | Para. 4.4.2.2 |
| 8. Disassembly and Inspection | Para. 4.4.14 |

4.2 MFJ DEVELOPMENT TESTS. Development tests for a full scale MFJ unit shall be based on the guidelines in MIL-STD-1522. These tests shall be used to verify compliance with the applicable requirements of this document.

The following development tests, as a minimum, shall be performed in the sequence shown below:

- | | |
|-----------------------------------|--------------|
| 1. Examination of Product | Para. 4.4.1 |
| 2. Proof Pressure Test | Para. 4.4.5 |
| 3. Environmental Temperature Test | Para. 4.4.9 |
| 4. Random Vibration Test | Para. 4.4.4 |
| 5. Mechanical Shock Test | Para. 4.4.10 |
| 6. Cleanliness | Para. 4.4.8 |
| 7. External Leakage Test | Para. 4.4.3 |
| 8. Functional Test | Para. 4.4.2 |
| 9. Thermal Cycling Test | Para. 4.4.11 |
| 10. Functional Test | Para. 4.4.2 |
| 11. Thermal Shock Test | Para. 4.4.12 |
| 12. External Leakage Test | Para. 4.4.3 |
| 13. Burst Pressure Test | Para. 4.4.6 |
| 14. External Leakage Test | Para. 4.4.3 |
| 15. Weight | Para. 4.4.7 |
| 16. Electrical Continuity Test | Para. 4.4.13 |
| 17. Disassembly and Inspection | Para. 4.4.14 |

4.3 Test Conditions

4.3.1 Test Procedure. Development test procedures shall be prepared and submitted to MSFC for approval at least 60 days prior to scheduled start of testing and shall include, but not be

limited to:

- (a) Specify how each test will be performed and sequence of operations and include the applicable test conditions such as temperature, pressure, test duration, environments, etc.
- (b) Specify test equipment to be used such as measuring instruments, recording apparatus, etc. Also specify the accuracy of the test equipment and frequency of calibration.
- (c) Specify the number of parts to be tested and list of tests to be performed on each part.
- (d) Designate the equipment axis where applicable.
- (e) Specify the interface diagram for all test instrumentation.
- (f) Specify pass or fail criteria.

4.3.2 Test Report. The development test results shall be documented in the form of a report which shall include, but not be limited to:

- (a) Configuration of the equipment tested.
- (b) Designation of equipment axis where applicable.
- (c) Specific required tests.
- (d) Required recording and measuring equipment.
- (e) Interface diagram for all test instrumentation.
- (f) Detail of test setup and sequence of operations for each test.
- (g) Measurements to be made before, during, and after each test.
- (h) Pass or fail criteria in terms of the performance measurements to be made, including the definition of allowable test measurement tolerances and corresponding design requirements parameter limits from this document.
- (i) Operating temperature ranges and temperature sensor locations.
- (j) Provisions to verify, if possible, satisfactory performance of redundant elements.

- (k) Specific environmental test conditions and instrumentation defined in detail (the values and tolerances of which shall comply with this specification).

4.3.3 Test Records. Records of all tests and inspection on the unit shall be made and maintained.

4.3.4 Test Tolerances. Where applicable, the maximum allowable tolerances (excluding instrument errors) for environmental test conditions shall be in accordance with MIL-STD-810 unless otherwise specified.

4.3.5 Standard Conditions. Unless otherwise specified herein, all measurements and tests shall be conducted within the following ambient conditions:

Temperature:	73 °F ± 8 °F
Relative Humidity:	30% minimum to 80% maximum
Pressure:	29 to 31 inches Hg

4.3.6 Accuracy of Measurements. The accuracy of the instrumentation and test equipment shall be verified in accordance with the requirements of MIL-C-45662.

4.4 Test Requirements

4.4.1 Examination of Product. The MFJ shall be visually inspected for conformance to envelope, and finish requirements. Welds shall be inspected in accordance with MSFC-STD-506.

4.4.2 Functional Test. The functional test shall consist of the following tests performed in the sequence given.

4.4.2.1 Pressure Drop Test. Water shall be flowed in the forward direction through the MFJ assembly at TBD lbm/sec with the MFJ at the design angular deflection of 13.0 degrees. The differential pressure across the MFJ assembly shall be measured and shall be TBD psid maximum with an inlet pressure 326 psig.

4.4.2.2 Angular Deflection Torque Test. TBD

4.4.3 External Leakage Test. Pressurize the inlet and outlet ports to 358 psig using gaseous helium (GHe) with the MFJ assembly at +70 °F. The helium leak rate from the MFJ assembly shall be measured and shall be 1×10^{-3} scc/sec maximum. Repeat the above test with MFJ assembly temperatures of -415 °F, -320 °F and +140 °F.

4.4.4 Random Vibration Test. Mount the MFJ assembly in the zero (null) angular deflection position on a support structure which simulates the stiffness of the flight support system of TBD.

Apply the vibration levels of 3.4.3.4. In addition, a 0.5 g (0 to peak) sinusoidal survey at 1 octave/min from 5 to 2000 Hz shall be conducted to identify major resonant frequencies and to verify the 650 Hz minimum natural frequency requirement during development testing. Repeat the above tests with the MFJ assembly at the design angular deflection position of 13.0 degrees. Refer to NASA TM-86538.

4.4.5 Proof Pressure Test. Pressurize the MFJ assembly with liquid nitrogen (LN_2) as specified in 3.2.1 (b).

The MFJ assembly shall be mounted in an unrestrained manner, including the flanged end connections. During pressurization, the rate of pressure increase shall not exceed 125 psig per minute. Hold the test proof pressure for a minimum of 1 minute. The pressure shall then be cycled as specified in 3.2.1 (b). After the completion of the pressure cycles, the pressure shall be relieved and the MFJ inspected for damage. The MFJ assembly shall show no evidence of damage or permanent deformation.

4.4.6 Burst Pressure Test. Pressurize the MFJ assembly with liquid nitrogen (LN_2) to the calculated burst pressure specified in 3.2.1 (c).

The MFJ assembly shall be mounted in an unrestrained manner, including the flanged end connections. During pressurization, the rate of pressure increase shall not exceed 125 psig per minute. The calculated burst pressure shall be maintained for at least 2 minutes. Under these conditions, structural failure shall not occur. The MFJ shall not be required to operate after exposure to the calculated burst pressure, but shall meet the external leakage requirements after the burst pressure test.

4.4.7 Weight. The MFJ dry weight shall be measured and shall not exceed 50 pounds.

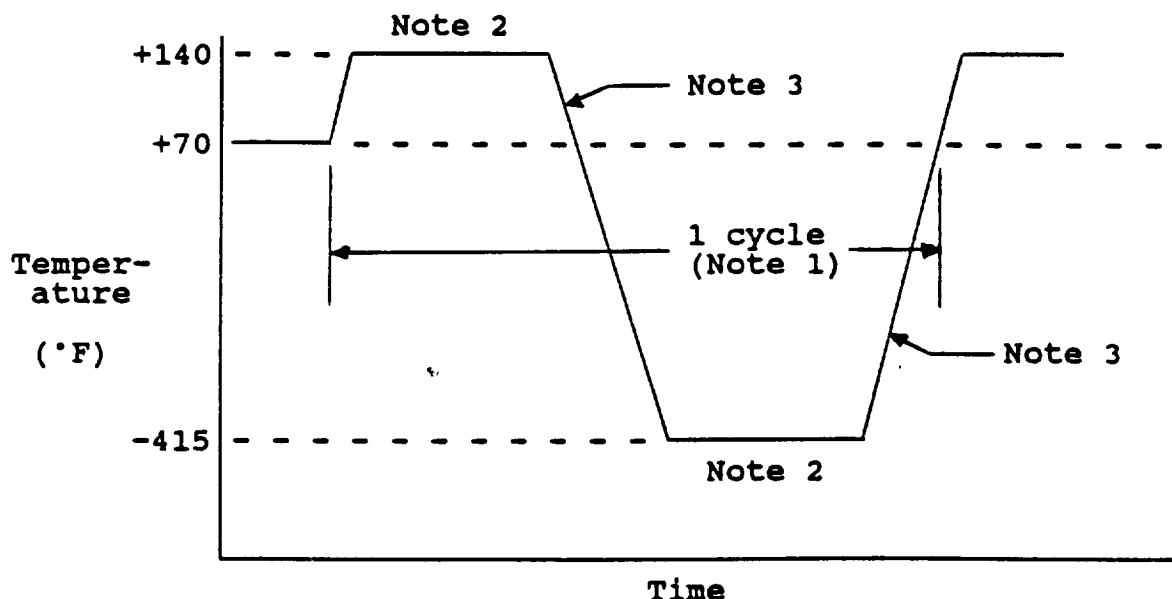
4.4.8 Cleanliness. The MFJ shall be subjected to the cleanliness tests of 3.7. Subsequent to the test, the MFJ shall be thoroughly dried in a vacuum oven with end flanges uncovered at a temperature not exceeding 135 °F.

4.4.9 Environmental Temperature Test. The MFJ, dry, shall be subjected to a temperature of -40 °F for 16 hours followed by a temperature of +140 °F for 8 hours at ambient pressure.

4.4.10 Mechanical Shock Test. The MFJ shall be subjected to the shock levels of 3.4.3.5 applied along each MFJ coordinate axis. The MFJ assembly shall be mounted on a support structure which simulates the stiffness of the flight support system of TBD.

4.4.11 Thermal Cycling Test. The thermal cycling test demonstrates that the MFJ assembly can withstand the expected environmental conditions. The MFJ shall be mounted in the chamber on a thermally controlled support plate. Reference temperature

sensors shall be mounted on the MFJ housing and shall be used to monitor and control the test temperatures. The test temperature conditions and cycle sequences are shown in Figure 4-1 for development testing.



- Notes:
1. Minimum of 5 cycles required.
 2. Duration as required for internal thermal equilibrium, but not less than 1.0 hour.
 3. Transitions between high and low temperatures shall be at an average rate of at least 2 °F per minute.

Figure 4-1 Temperature Cycling for Development Test

4.4.12 Thermal Shock Test. The requirements are TBD for testing the MFJ in both the horizontal and vertical orientations.

4.4.13 Electrical Continuity Test. Measure the electrical continuity from MFJ inlet to MFJ outlet. The MFJ shall be electrically continuous per the requirements of 3.3.6.

4.4.14 Disassembly and Inspection. After completion of the design burst pressure test, the MFJ shall be disassembled and inspected for any signs of wear or damage and to determine the integrity of weld joints. Wear or damage shall be documented. Inspection for wear of dynamic sealing surfaces shall be

Revision 0
July 31, 1991

accomplished with a Surface Finish Analyzer which meets the requirements of A-A-50767 (suitable unit is manufactured by Marduth Products, Model SR-14B). Weld joints shall be cross-sectioned and inspected for weld penetration, voids, and quality. Photographs of weld sections shall be documented in the development test report.

APPENDIX B

STATIC SEALS TEST PLAN



**National Aeronautics and
Space Administration**

**George C. Marshall Space Flight Center
Science and Engineering
Contract NAS 8-37814**

Test Plan for the Low Temperature Testing of Static Seals

Control Mechanisms & Propellant Delivery Branch

Report No.: 324-006-91-006

Prepared by:

Lila F. Pegram

December 1991



**Sverdrup Technology, Inc.
MSFC Group
620 Discovery Drive
Huntsville, Alabama 35806**

Test Plan for the Low Temperature Testing of Static Seals

Prepared by: Lila F. Pegram Date: 12/3/91
Lila F. Pegram, Propulsion Engineer
Sverdrup Technology, Inc.
Propulsion Department, Component Development

Approved by: Gary W. Lees Date: 12-3-91
Gary Lees, Task Leader
Sverdrup Technology, Inc.
Propulsion Department, Component Development

Approved by: Tom Beasley Date: 12-3-91
Tom Beasley, MP64 Supervisor
Sverdrup Technology, Inc.
Propulsion Department, Component Development

Approved by: Sharon Seay Date: 12/18/91
Sharon Seay, Safety
Sverdrup Technology, Inc.

Approved by: Ken Farrow Date: 12/5/91
Ken Farrow, Director
Sverdrup Technology, Inc.
Propulsion Department

Approved by: John Harbison Date: 2/28/92
John Harbison, Chief
Control Mechanisms & Propellant Delivery Branch
Component Development Division
Propulsion Laboratory
NASA/Marshall Space Flight Center

Approved by: Charles S. Cornelius Date: 2-28-92
Charles S. Cornelius, Chief
Component Development Division
Propulsion Laboratory
NASA/Marshall Space Flight Center

ABSTRACT

The requirements and plan of test for the Low Temperature Testing of Static Seals to be conducted in Bldg. 4656 at Marshall Space Flight Center (MSFC) are presented in this document. A description of the test configuration, subassemblies and components, and instrumentation requirements are included. This test plan shall provide the requirements for measuring leakage of static seals when exposed to extreme temperatures. The test plan identifies test requirements as well as instrumentation, test location, data products, and safety considerations.

Successful completion of these tests shall provide a database of the properties of ethylene propylene, silicone, and Viton® when exposed to low temperatures.

KEY WORDS

Ethylene Propylene

Fluorocarbon

Gask-O-seal

Helium Mass Spectrometer

Low Temperature

O-ring Seal

Silicone

Static Seals

Test Plan

Viton®

TABLE OF CONTENTS

SECTION	PAGE
ABSTRACT	ii
KEY WORDS	iii
TABLE OF CONTENTS	iv
LIST OF FIGURES	v
ABBREVIATIONS AND ACRONYMS	vi
1.0 INTRODUCTION	1
1.1 BACKGROUND	1
1.2 PURPOSE	1
1.3 SCOPE	2
1.4 APPLICABLE DOCUMENTS	2
2.0 SUMMARY	3
2.1 TEST OBJECTIVES	3
2.2 TEST DESCRIPTION	3
2.3 TEST HARDWARE	3
2.3.1 LN2 Dewar	3
2.3.2 Helium Mass Spectrometer	4
2.3.3 LN2 Bath	4
2.3.4 Helium Source	4
2.3.5 Test Article and Housing	4
2.3.6 Heater Elements	4
2.3.7 Cryogenic Valves	5
3.0 TEST REQUIREMENTS	8
3.1 TEST CONDITIONS	8
3.2 INSTRUMENTATION SYSTEM	8
3.2.1 Data Acquisition	8
3.3 FACILITIES	8
3.4 PHOTOGRAPHY	8
4.0 TEST CONDUCT	10
4.1 TEST MATRIX	10
4.2 TEST CONSTRAINTS	11
5.0 DATA ANALYSIS AND FINAL TEST REPORT	12
5.1 DATA HANDLING PLAN	12
5.2 QUICK LOOK DATA ANALYSIS	12
5.3 DATA ANALYSIS AND FINAL TEST REPORT	12
6.0 TEST IMPLEMENTATION AND CONTROL	13
6.1 CONFIGURATION CONTROL	13
6.2 TEST CONTROL	13
6.3 TEST READINESS REVIEW	13
6.4 SAFETY	14
6.5 TEST PROCEDURES	14

LIST OF FIGURES

FIGURE		PAGE
1	TEST SETUP IN 4656	5
2	TEST SCHEMATIC	6
3	TEST ARTICLE AND HOUSING	7
4	BLDG 4656 FLOOR PLAN	9

ABBREVIATIONS AND ACRONYMS

GHe	Gaseous Helium
HMS	Helium Mass Spectrometer
ICD	Interface Control Document
LN ₂	Liquid Nitrogen
MSFC	Marshall Space Flight Center
NASA	National Aeronautics and Space Administration
O ₂	Oxygen
RTLS	Return to Launch Site
SSME	Space Shuttle Main Engine
TM	Technical Memorandum
TRR	Test Readiness Review

1.0 INTRODUCTION

1.1 BACKGROUND

The Viton® elastomer Gask-O-seals used in the helium systems of the SSME are designed to operate for the temperature range between -40°F and 700°F. The material property of this fluorocarbon seal is satisfactory during normal Space Shuttle flight conditions. It has been noted that the minimum temperature requirement (-40°F) of the material can be violated in special cases during re-entry (-70°F) or during an RTLS abort (-80°F). Under these low temperature extremes the Viton® Gask-O-seal may lose its ability to perform as required. Currently this condition is covered for SSME by an ICD change which increased the allowable seal leakage rate. Static seal manufacturer data provides little information of the material characteristics of seals when exposed to low temperatures.

A literary search of seal materials was performed to determine some candidate alternate materials for testing. In addition to the Gask-O-seal, the materials chosen to be tested are ethylene propylene, fluorocarbon, and silicone.

Listed below are some of the characteristics of these materials:

Ethylene Propylene - Ethylene propylene rubber has a useful temperature range that extends from -70 to 300°F. The ethylene propylene compound to be tested is E692-75, which has a gas permeability rate of approximately 19.7×10^{-8} std cc cm/cm² sec bar at room temperature.

Fluorocarbon (Viton®) - Sealing with fluorocarbon is difficult at temperatures below -20°F unless a special low temperature type is used that seals to -40°F. The gas permeability rate of fluorocarbon sealing a helium fluid is approximately 12.7×10^{-8} std cc cm/cm² sec bar. The fluorocarbon compounds to be tested is V835-75 (modified fluorocarbon Viton).

Silicone - Silicone rubber has a wide temperature range. For instance, some silicone compounds remain flexible below -175°F, and some resist temperatures to 700°F for short periods. The temperature range for standard, off-the-shelf silicone is -65 to 350°F, however, compound S383-70 may be used to reach temperatures of -175°F or lower. The gas permeability rate of silicone rubber at room temperature is 263×10^{-8} std cc cm/cm² sec bar.

1.2 PURPOSE

The purpose of this test series is to develop a database of the properties of static seal materials when exposed to low temperatures. Information contained in this database shall provide a resource for the determination of candidate materials in future low temperature seal applications.

1.3 SCOPE

This document presents the test plans and test conditions for the low temperature testing of static seals to be used in helium systems. The test equipment necessary for this test is identified in section 2.3.

1.4 APPLICABLE DOCUMENTS

The following documents of the issue in effect on the date of this document form a part of these plans to the extent specified herein. In the event of conflict between documents referenced below and other detail content of this document, the requirements specified herein shall govern. Safety documents are an exception to this rule; they take precedence over other requirements.

SPECIFICATIONS

NASA

MSFC-SPEC-164A	Cleanliness of Components for use in OXYGEN, FUEL, and PNEUMATIC Systems
MSFC-SPEC-522	Design Criteria for Controlling Stress Corrosion Cracking

STANDARDS

Military

MIL-STD-413	Visual Inspection Guide for Rubber O-rings
MIL-STD-810	Environmental Test Method and Engineering Guidelines

NASA

MSFC-STD-555	MSFC Engineering Documentation Standard
--------------	---

Industry

ASTM D1414	Standard Test Method for Rubber O-rings
SAE MAP 3439	O-ring Groove Design

MANUALS

NASA

MMI-1700.4	Safety and Environmental Health Standards
MMI-1710.1	Safety Review and Approval of Potential Hazardous Facilities and Activities

2.0 SUMMARY

2.1 TEST OBJECTIVES

It is the objectives of the testing outlined in this document, to determine the actual performance of the Viton® Gask-O-seals when exposed to temperature conditions from 0°F to -150°F. Additional testing shall be conducted on static o-ring seals of several materials to develop a database for use in future seal applications.

Specific areas to be addressed in accomplishing these objectives are:

- Develop, through detailed instrumentation, a better understanding of seal operating characteristics when exposed to extreme low temperatures.
- Provide needed data to validate existing and new analytical models and provide a resource for the determination of candidate materials for use in future low temperature seal applications.
- Evaluate the performance of these seals at varying compression levels.

2.2 TEST DESCRIPTION

The seal configuration shall consist of a test article which is contained between two bolted flanges located inside of a sealed housing (Figure 3). The inside surface of the test article shall be pressurized to 850 psig with GHe. The outside of the test article shall be held in a vacuum. The housing is placed inside of a open dewar which is filled with LN₂ to a level two inches over the top of the housing. Kapton heaters which are mounted inside the housing are used to vary the temperature of the test article between ambient and -150°F.

2.3 TEST HARDWARE

Figures 1 and 2 show the test setup and schematic of test hardware. Major components are described in the following sections.

2.3.1 LN₂ Dewar

An LN₂ Dewar shall be used to supply the LN₂ needed to chill the test article. The dewar shall have a minimum capacity of 100 liters. It shall be insulated for storing LN₂ for up to 5 days and shall have a built-in pressure relief device. The LN₂ supply line to the bath shall be 1" stainless steel tubing and shall be insulated to maintain temperature of the LN₂.

2.3.2 Helium Mass Spectrometer

The Helium Mass Spectrometer (HMS) shall be a Veeco, Model No. MS-170, or equivalent. The HMS shall be used to measure the amount of helium leakage through the seal. It shall provide automatic readout from 10 atm cc/sec (1.15×10^4 sccm Helium) to 6×10^{-11} atm cc/sec (air) (1.0×10^{-8} sccm Helium) with quantitative gross leak data. The response time is less than 1.5 seconds in the fine leak mode (10^{-10} to 10^{-5}) and less than 2.5 seconds in the gross leak model (10^{-4} to 10). The leak range shall be between 10 atm cc/sec to 6×10^{-11} atm cc/sec for air at full pumping speeds, presented over 11 ranges with automatic or manual range selection. Larger leaks can be measured using throttling techniques. Power requirements with a 10.6 cfm pump are 30 amps at 115 volts, 60 Hz. This unit shall include a mechanical forepump and a roughing pump.

2.3.3 LN₂ Bath

The LN₂ Bath shall consist of an open top dewar and shall have a minimum capacity of 25.5 liters. It is required to chill the test article to the desired temperature. A styrofoam lid shall be used over the bath to minimize boil-off of the LN₂.

2.3.4 Helium Source

The test article shall be pressurized to 850 psig with Helium gas. The Helium supply shall consist of a K-bottle, pressurized regulator, relief valve, and a dump valve.

2.3.5 Test Article and Housing

There shall be two seal configurations tested. The first test article shall be a Viton® Gask-O-Seal in the configuration currently used in the SSME Helium System. The second test article shall be a static o-ring face seal configuration. The o-rings shall conform to Parker size no. 2-022 (0.989 ± 0.010 O.D., 0.070 ± 0.003 cross sectional diameter).

The design and dimensions of the test housing are contained in drawings 96M66990, 96M66991, and 96M66992.

2.3.6 Heater Elements

Kapton heaters shall be used to control the temperature of the test article. The heaters shall be 1.5" long x 2.35" wide x 0.007" thick and have a minimum operating temperature of -319°F. The heater element shall require AC power and shall provide a maximum operating watt density of 3 watts/in². The power leads shall consist of 18 AWG wire which shall be mounted on the 1.5" side of the heater element. The heaters shall be attached to the test housing using epoxy adhesives.

The heater controller shall be a Watlow, Model No. 945 Digital Controller, or equivalent. The controller shall monitor temperature of the test article and adjust heater power input to maintain test settings. The controller shall provide automatic cold junction compensation for thermocouples, and shall be compatible with type K thermocouples. Power requirements shall be 120 VAC.

2.3.7 Cryogenic Valves

Two cryogenic valves shall be required for the LN₂ supply and dump lines. These valves must withstand temperatures of -320°F.

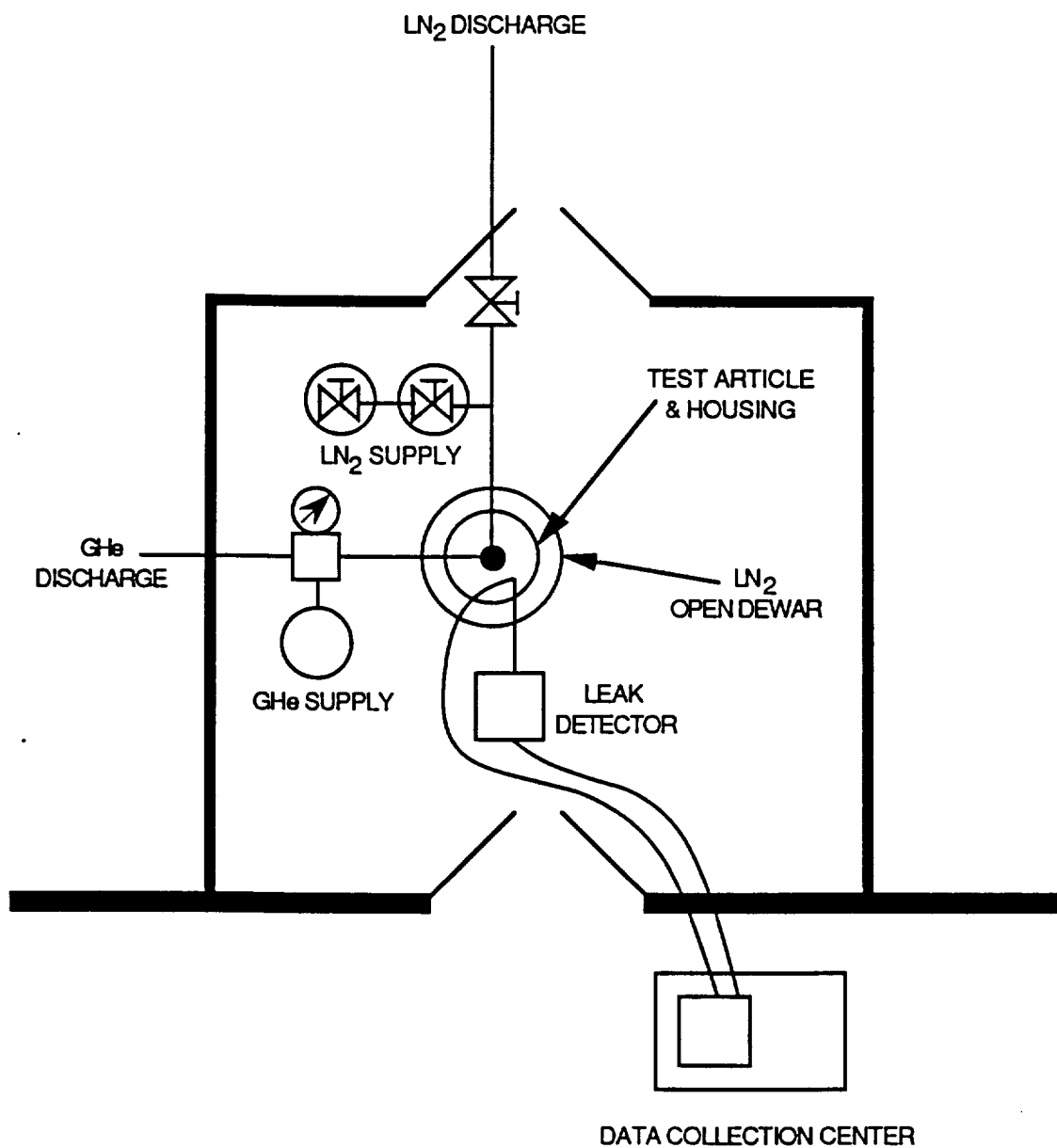


FIGURE 1. TEST SETUP IN 4656

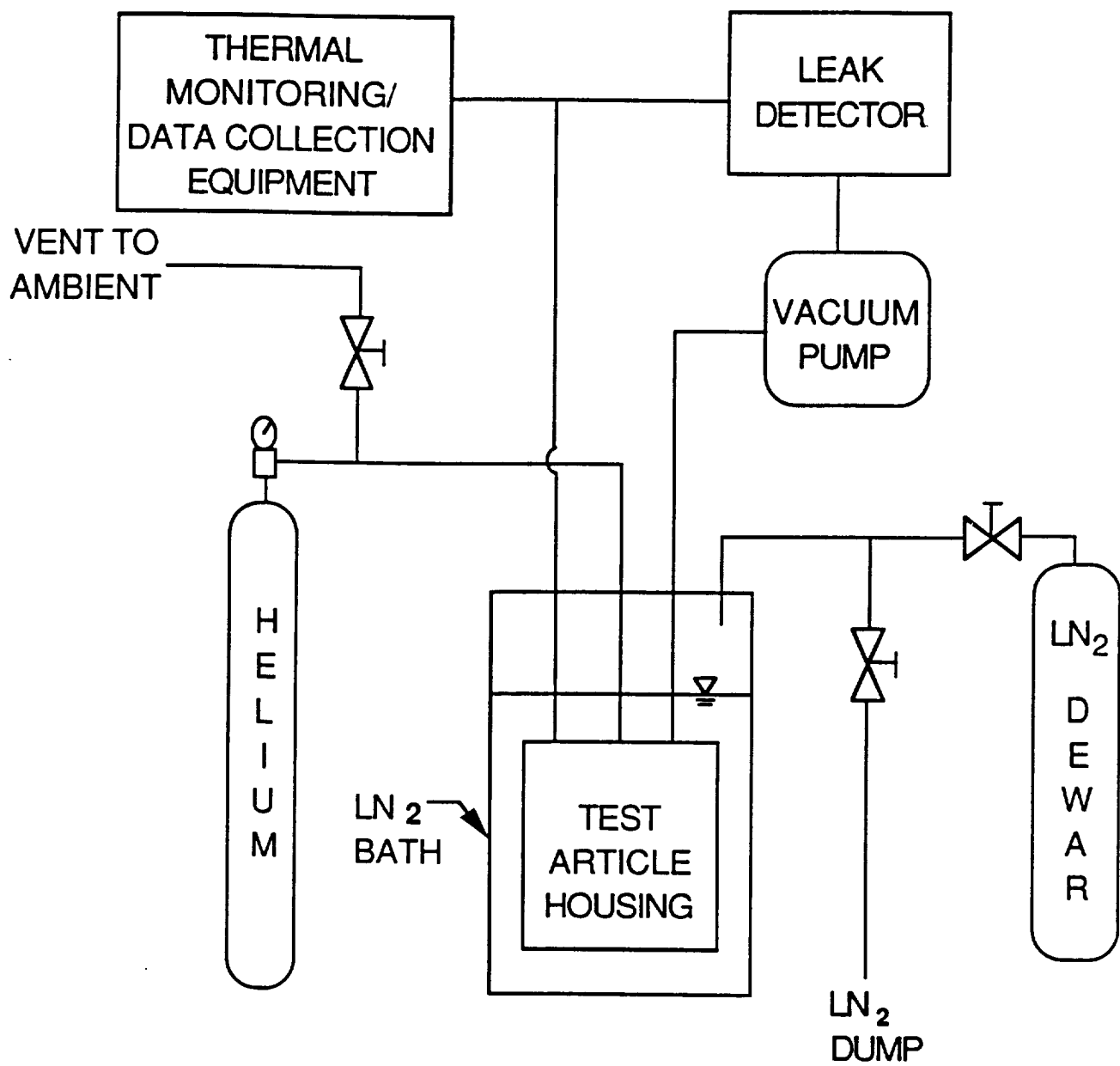


FIGURE 2. TEST SCHEMATIC

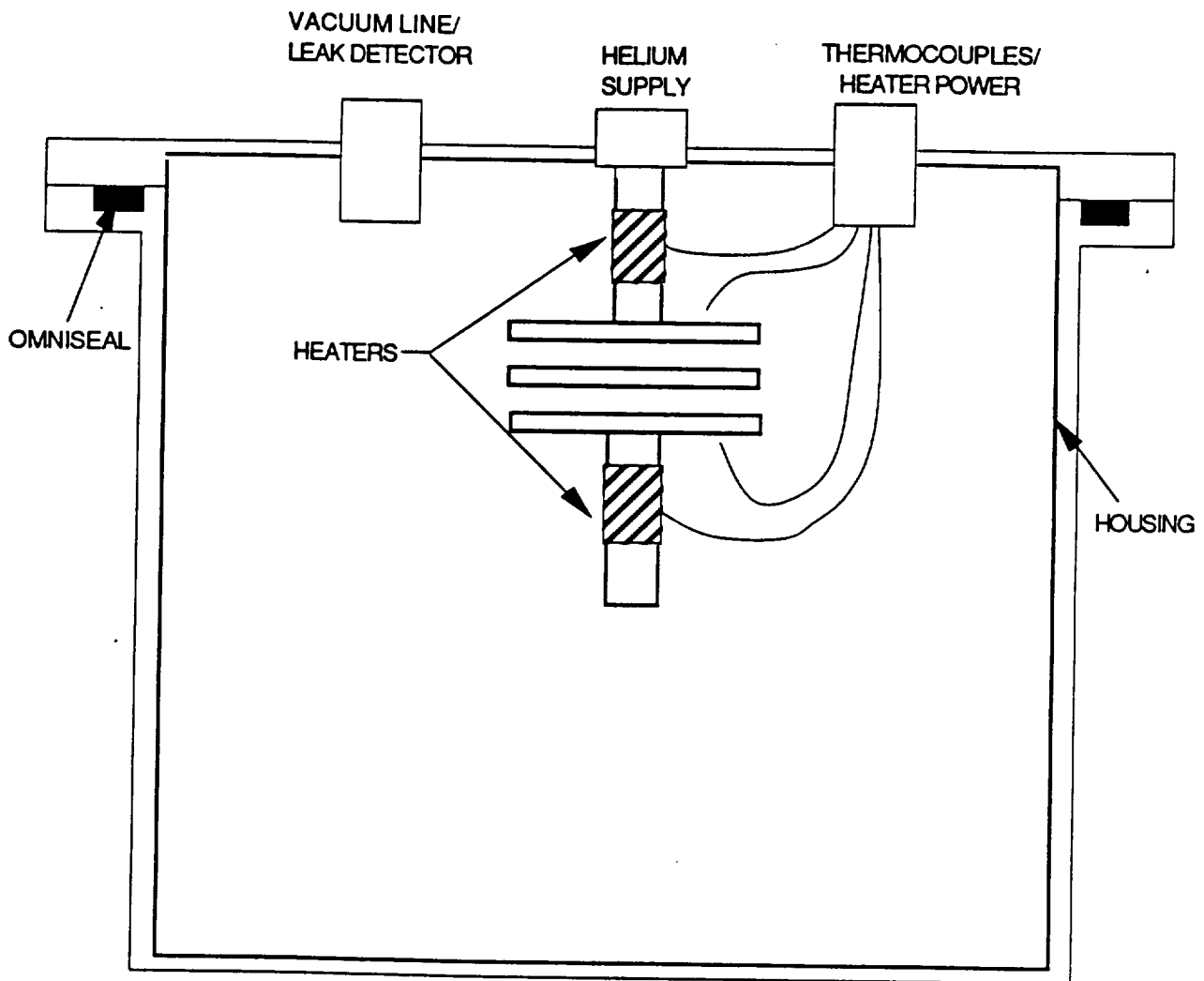


FIGURE 3. TEST ARTICLE AND HOUSING

3.0 TEST REQUIREMENTS

3.1 TEST CONDITIONS

The test article and housing shall be placed in the empty LN₂ bath and LN₂ shall be added slowly to decrease the temperature. The heater controller shall be used to maintain the desired temperature.

A pre- and post-test inspection of the seals shall be performed to document the condition of the seal. During pre- and post-test inspections photographs shall be taken of all seals.

3.2 INSTRUMENTATION SYSTEM

3.2.1 Data Acquisition

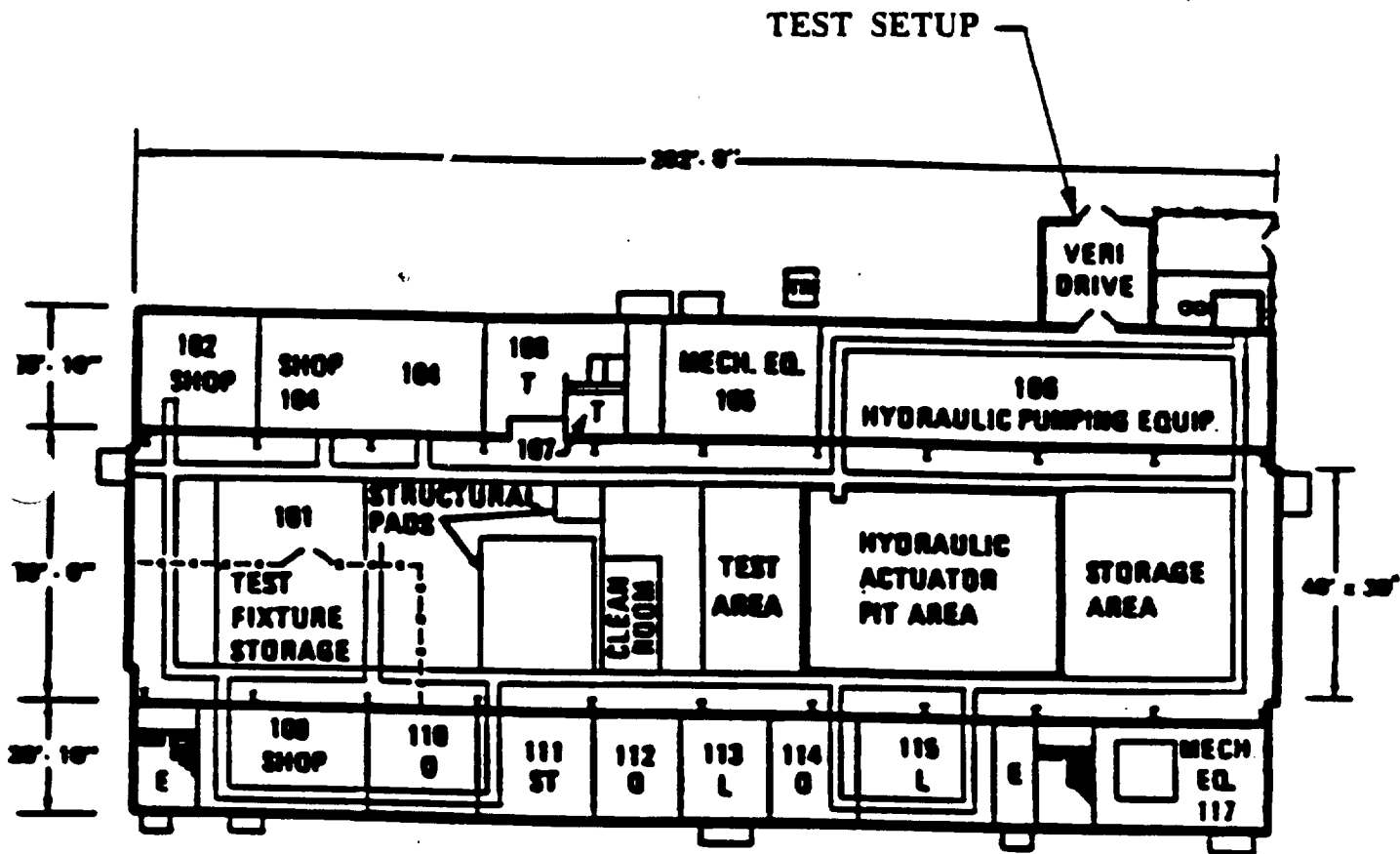
The data acquisition system shall be provided by MSFC and shall include an HP-85P data logger and and HP 3054A data acquisition/control unit. The system must as a minimum provide the following: a plug-in interface to an IBM PC XT, AT or compatible computer; accept analog or digital signals including thermocouples, millivolts, volts, and milliamp current signals including 8 analog input channels and 8 digital I/O channels; must include software which shall support graphical and tabular display of real-time data, control of alarm conditions, datalogging to printer or disk, and export data to most spreadsheet programs for further analysis.

3.3 FACILITIES

The facility to be used to conduct this test is located in the southeast corner of Building 4656, MSFC, Huntsville, Alabama. (See Figure 4)

3.4 PHOTOGRAPHY

Still photographs shall be required to record pretest configurations and post test events. All film and prints must be properly annotated and classification labeled. The date and test run number shall be included in the picture where possible.



FIRST FLOOR PLAN

*NOTE: NORTH DOOR SAME AS SOUTH.

FIGURE 4. BLDG 4656 FLOOR PLAN

4.0 TEST CONDUCT

4.1 TEST MATRIX

The test requirements for each test run are shown in Table I. The tests are to be accomplished in the run order specified. The test order may be changed at the direction of the Test Conductor as conditions change due to test results, test anomalies, or test article failures.

Four Gask-O-seals shall be tested first and the amount of compression applied shall not be varied. The seal compression of the o-ring seals shall be varied to determine if an increase in squeeze shall improve the ability to maintain a seal at low temperatures. This compression shall be varied using shims. Three o-ring seal materials were chosen to be tested and four o-rings for each material and each compression level shall be tested. The squeeze for the static seal o-rings tested shall be varied from 10% to 40% in 10% increments. This makes the total number of seals to be tested equal to 52.

SEAL TYPE	MATERIAL	TOTAL # OF SEALS	# OF SEALS AT EACH COMPRESSION LEVEL			
			10%	20%	30%	40%
GASK-O-SEAL	VITON	4	N/A	N/A	N/A	N/A
O-RING	VITON V835-75	16	4	4	4	4
O-RING	ETHYLENE PROPYLENE E692-75	16	4	4	4	4
O-RING	SILICONE S383-70	16	4	4	4	4

Table I. Test Requirements

The test cycle shall be initiated after the test article has been thermally soaked for a minimum of ten minutes at 0°F. The temperature shall be reduced in 25° increments $\pm 10^\circ$ until the test article reaches -100°F. The temperature shall then be reduced in 10° increments $\pm 5^\circ$ until the test article reaches -150°F. The test article temperature shall then be increased in 25° increments until the test article reaches 0°F. Helium leakage rate shall be measured continuously throughout the cycle.

4.2 TEST CONSTRAINTS

Prior to testing the following conditions must be met:

- A. Test readiness review completed and all action items must be closed.
- B. Instrumentation must be calibrated and fully functional as required per section 3.2.
- C. Pretest configuration photography must be complete as required in section 3.6.
- D. All personnel shall be at their assigned station as required by the Test Conductor.
- E. All personnel shall have attended a safety briefing and have read the safety procedures.
- F. All safety warning equipment and indicators shall be in place and functioning.
- G. Safety Review per MMI-1710.1.

5.0 DATA ANALYSIS AND FINAL TEST REPORT

5.1 DATA HANDLING PLAN

All test data shall be recorded on disc or magnetic tape. The raw data and engineering unit data tapes from the NASA data acquisition system shall be provided to Sverdrup. The data tapes, photographs, and film shall be retained by the Sverdrup organization and identified as to name, test run, test date, and test site. NASA shall retain copies of the data records.

5.2 QUICK LOOK DATA ANALYSIS

Efficient and safe conduct of the test requires that real time and quick look data be available during test. Real time monitoring of test parameters is required during the test.

5.3 DATA ANALYSIS AND FINAL TEST REPORT

The test analysis shall be performed by Sverdrup personnel and shall be incorporated into a database for future reference.

The test data shall contain test report logs as defined in the following. A complete test log shall be maintained by the test personnel and contain all information regarding testing operations. A final report shall be prepared and presented in the NASA TM format and shall include the following:

- A. Summary of test activities
- B. Description of the test set up
- C. As-run test procedures
- D. A listing and description of failures, anomalies, test procedure deviations, and data losses
- E. Results of testing
- F. Detailed test data
 - (1) Test data
 - (2) Photographs
 - (3) Timeline of test
 - (4) Log book of events
 - (5) Equipment list with calibration dates.

6.0 TEST IMPLEMENTATION AND CONTROL

6.1 CONFIGURATION CONTROL

The test article, test fixture, etc. shall be controlled by drawings used to manufacture each item. Any changes to the drawings, assembly of test units, test plan, and test procedures shall be documented by the Test Conductor using redlines with concurrence from MSFC/EP64.

6.2 TEST CONTROL

Sverdrup personnel shall act as the Test Conductor. The Test Conductor shall direct test activities per approved test procedures. A running log shall be kept of all test activities, test setup, problems encountered and solutions, and any unplanned events.

Any changes to the test procedures must be approved by the Test Conductor before implementation.

6.3 TEST READINESS REVIEW

Test Readiness Review (TRR) meeting shall be held prior to the initial test to review test objectives with the test team. At the TRR the following items shall be reviewed:

- A. Test article and facility readiness
- B. Mandatory measurement list
- C. Quick-look measurement list
- D. Instrumentation readiness
- E. Photographic requirements
- F. Data acquisition and reduction system readiness
- G. Personnel assignments
- H. Test and safety procedures

The TRR may identify open items that require action prior to testing. All open items affecting the test must be completed prior to test start.

APPENDIX C

TORQUE AND BALL RADIUS EQUATION PROGRAMS

THIS PROGRAM IS FOR MR. BILL COOLEY
 THE PROGRAM IS INTENDED TO FIND A BALL RADIUS TO PIPE RADIUS
 OF A TYPICAL MFJ, A SINGLE BALL JOINT, AND A DUAL BALL JOINT.
 IT'S METHOD IS TO INCREMENT RB ONLY

CLS

OPEN "C:\DOS\BALLJOI2.DAT" FOR OUTPUT AS #2

PS = "####.#### ####.#### ####.#### ## ## ###.##"

FS = "##### ##### ##### #####"

'OMEGA = ARC LENGTH OF BALL

RP = 2.6 'PIPE RADIUS

'RB = BALL RADIUS

WS = 1! 'ARC LENGTH OF SEAL

WB = 1! 'ARC LENGTH OF BEARING

WJ = 1.5 'ARC LENGTH OF JOINT

B1 = 13.5 'POSITIVE FLEX ANGLE, (DEGREES)

B2 = 13.5 'NEGATIVE FLEX ANGLE, (DEGREES)

PI = 3.14159

'RBRG = RADIUS OF THE BEARING

TBAG = .2 'THICKNESS OF BEARING

MU = .1 'FRICTION COEFFICIENT

PD = 343 'DESIGN PRESSURE, PSIA

RB = 1000

DELRB = 1000

COUNT = 0

'***** CALCULATION *****

10 PHI = ATN((RP / RB) / SQR(1 - (RP / RB) ^ 2))

OMEGA1 = PI / 2 - PHI

OMEGA2 = (((B1 + B2) / 57.2958) * 2) + ((WS + WB + WJ) / RB)

IF COUNT = 1000 THEN

GOTO 20

END IF

IF OMEGA1 = OMEGA2 OR ABS(OMEGA1 - OMEGA2) <= .0001 THEN

RBRP = RB / RP

RBRG = RB

BS = (WS / 2 + WJ) / RB + B2 * PI / 180

BBRG = (WB / 2 + WS + WJ) / RB + B1 * PI / 180 + 2 * B2 * PI / 180

PRINT #2, ., "TYPICAL SINGLE BALL"

PRINT #2, " RB RP RB/RP OMEGA(DEG.) PHI(DEG.) BBRG"

PRINT #2, USING PS; RB; RP; RBRP; OMEGA2 * 180 / PI; PHI * 180 / PI; BBRG * 180 / PI

PRINT #2, ""

'FRICTION FORCES

RSC = RB * COS(BS)

ASCC = PI * RSC ^ 2

FA = PD * ASCC

FFN = FA / SIN(BBRG)

FF = MU * FFN

'ROTATION TORQUE

TR = (MU * RBRG * PD * RSC ^ 2) / (3.8197 * TAN(BBRG))

'HINGE TORQUE

TF = (MU * RBRG * PD * RSC ^ 2) / (9 * SIN(BBRG))

PRINT #2, "F(AXIAL) F(NORMAL) F(FRICTIONAL) T(ROTATIONAL) T(FLEX)"

PRINT #2, USING FS; FA; FFN; FF; TR; TF

PRINT #2, ""

PRINT #2, ""

ELSEIF OMEGA2 < OMEGA1 THEN


```

    RB = RB - DELRB
    DELRB = DELRB / 1.5
    RB = RB + DELRB
    COUNT = COUNT + 1
    GOTO 10
ELSE
    RB = RB + DELRB
    COUNT = COUNT + 1
    GOTO 10
END IF
*****
20 RB = 1000
DELRB = 1000
COUNT = 0
30 PHI = ATN((RP / RB) / SQR(1 - (RP / RB) ^ 2))
OMEGA1 = PI / 2 - PHI
OMEGA2 = ((B1 + B2) / 57.2958) + ((WS + WJ) / RB)
    IF COUNT = 1000 THEN
        GOTO 40
    END IF
    IF OMEGA1 = OMEGA2 OR ABS(OMEGA1 - OMEGA2) <= .0001 THEN
        RBRP = RB / RP
        RBRG = RB + TBAG
        BBRG = (WS / 2 + WJ) / RB + B2 * PI / 180
        BBRG = BS
        PRINT #2, , , "SINGLE CONCENTRIC"
        PRINT #2, "    RB    RP    RB/RP OMEGA(DEG.)  PHI(DEG.)  BBRG"
        PRINT #2, USING P$; RB; RP; RBRP; OMEGA2 * 180 / PI; PHI * 180 / PI; BBRG * 180 / PI
        PRINT #2, ""
    FRICITION FORCES
        RSC = RB * COS(BS)
        ASCC = PI * RSC ^ 2
        FA = PD * ASCC
        FFN = FA / SIN(BBRG)
        FF = MU * FFN
    ROTATION TORQUE
        TR = (MU * RBRG * PD * RSC ^ 2) / (3.8197 * TAN(BBRG))
    HINGE TORQUE
        TF = (MU * RBRG * PD * RSC ^ 2) / (9 * SIN(BBRG))
        PRINT #2, "F(AXIAL)  F(NORMAL)  F(FRICTIONAL)  T(ROTATIONAL)  T(FLEX)"
        PRINT #2, USING F$; FA; FFN; FF; TR; TF
        PRINT #2, ""
        PRINT #2, ""
    ELSEIF OMEGA2 < OMEGA1 THEN
        RB = RB - DELRB
        DELRB = DELRB / 1.5
        RB = RB + DELRB
        COUNT = COUNT + 1
        GOTO 30
    ELSE
        RB = RB + DELRB
        COUNT = COUNT + 1
        GOTO 30
    END IF
*****
40 RB = 1000
DELRB = 1000
COUNT = 0

```

```

50 PHI = ATN(RP / RB) / SQR(1 - (RP / RB) ^ 2)
OMEGA1 = PI / 2 - PHI
OMEGA2 = ((B1 + B2) / (57.2958 * 2)) + ((WS + WJ) / RB)
  IF COUNT = 1000 THEN
    END
  END IF
  IF OMEGA1 = OMEGA2 OR ABS(OMEGA1 - OMEGA2) <= .0001 THEN
    RBRP = RB / RP
    RBRG = RB + TBAG
    BBRG = (WS / 2 + WJ) / RB + B2 * PI / 180
    BBRG = BS
    PRINT #2, , , "DUAL CONCENTRIC"
    PRINT #2, "  RB      RP      RB/RP  OMEGA(DEG.)  PHI(DEG.)  BBRG"
    PRINT #2, USING P$; RB; RP; RBRP; OMEGA2 * 180 / PI; PHI * 180 / PI; BBRG * 180 / PI
    PRINT #2, ""
  FRICITION FORCES
    RSC = RB * COS(BS)
    ASCC = PI * RSC ^ 2
    FA = PD * ASCC
    FFN = FA / SIN(BBRG)
    FF = MU * FFN
  ROTATION TORQUE
    TR = (MU * RBRG * PD * RSC ^ 2) / (3.8197 * TAN(BBRG))
  HINGE TORQUE
    TF = (MU * RBRG * PD * RSC ^ 2) / (9 * SIN(BBRG))
    PRINT #2, "F(AXIAL)  F(NORMAL)  F(FRICTIONAL)  T(ROTATIONAL)  T(FLEX)"
    PRINT #2, USING F$; FA; FFN; FF; TR; TF
  ELSEIF OMEGA2 < OMEGA1 THEN
    RB = RB - DELRB
    DELRB = DELRB / 1.5
    RB = RB + DELRB
    COUNT = COUNT + 1
    GOTO 50
  ELSE
    RB = RB + DELRB
    COUNT = COUNT + 1
    GOTO 50
  END IF
CLOSE #2
END

```

APPENDIX D
(See Seperate Binding)

**DYNAMIC SEAL TEST FIXTURE
PRELIMINARY DESIGN REVIEW DRAWING PACKAGE**

96M66993-1 MFJ SEAL TEST FIXTURE, SHEETS 1-10
96M66994-1 TEST FIXTURE STRONG BACK, SHEETS 1-4
96M66995-1 ACTUATOR INTERFACE, SHEETS 1-2
96M66996-1 FINAL ASSEMBLY